CRANFIELD UNIVERSITY

GEOFFREY B. JONES

PERFORMANCE MODELLING OF WINDMILLING GAS TURBINES

SCHOOL OF ENGINEERING

Eng.D. Thesis

CRANFIELD UNIVERSITY

SCHOOL OF ENGINEERING

Eng.D. Thesis

1998-2002

GEOFFREY B. JONES

Performance Modelling of Windmilling Gas Turbines

Supervisors:	Prof. P. Pilidis (School of Engineering),
	Prof. A. Harrison (School of Management),
Industrial Contact:	Mr. A. Brook (Rolls-Royce plc)

4 October 2002

This thesis is submitted in partial fulfilment of the requirements for the Degree of Engineering Doctorate.

Abstract

This thesis presents work carried out with the aim of improving the modelling of windmilling in gas turbine performance. The work also examines the phenomenon of relight.

Methods of representing the performance of the turbomachinery components are investigated and recommendations are made for the use of a number of non-dimensional variables on which to map turbomachinery performance.

A performance model was built using the turbomachinery characteristics proposed in order to test the robustness of the representations. The model was written in Fortran 90 and coupled directly to a spreadsheet package to allow easy analysis of the results. The effect of choice of representation, in terms of both the robustness and the results obtained, is presented.

Techniques for the extrapolation of above-idle turbomachinery characteristics to the sub-idle region for starting and windmilling modelling are examined. A number of techniques are analysed and two new methods are proposed. These are the use of computational fluid dynamics to generate a zero speed torque and pressure loss relationship and the use of a semi-empirical stage-by-stage extrapolation method.

Some aspects of relight combustion, particularly in respect of ignition, are studied. Also investigated were the accessory systems acting on the gearbox of the engine.

An assessment of the design process and the consequent benefits to the company of improved modelling were presented, in terms of the reduction in design risks and in project development costs. It was shown that research of this nature can have a substantial impact on an engine's development programme

Throughout the doctoral programme, Masters and exchange students were used to broaden the scope and depth of the research. This thesis examines how best to ensure that the results of such collaboration are positive and presents the approach which this researcher used.

Table of Contents

List of Figures

List of Tables

Acknowledgements

This work has been funded jointly by the Engineering and Physical Sciences Research Council and Rolls-Royce plc and so my thanks must be extended to those two bodies for allowing this research to be undertaken.

In any work of this magnitude, there are many more people responsible for it than just the named author. Over the four years that it has taken to write this thesis, I've been helped and supported by a wide variety of people: academics, Rolls-Royce staff, M.Sc. and exchange students, family and friends.

Naturally, the most direct support has come from my main academic supervisor, Prof. Pilidis. Perhaps one of the more important things he kept reminding me of, however, is the value of a holiday. Then, I've also seen a lot of help from Prof. Harrison, Mr. Jackson, Dr. Ramsden and Prof. Singh.

Within Rolls-Royce, I have been provided with help and supervision from Barry Curnock, John Martin, Anthony Brook, Dave Fillingham, Richard Tunstall, Nick Merriman, Roy Meads, Roy Curry and Arthur Rowe. Particular recognition must be given to Barry Curnock, whose supervision through the earlier stages of the work and whose advice throughout the life of the project have proven very useful.

I have been lucky while pursuing my doctorate in that I have had the help of a number of MSc and SOCRATES students. So, thank you very much to Sog-Kyun, Cyrus, Mathieu, Vicent, Joan Josep, David, Omar, Romain and Chris.

Without the support of my parents and my brother, undertaking this doctorate would have been impossible. They have been there when I have needed them.

Finally, I would like to say thank you to the friends who have helped me while at Cranfield. Within the UTC, I've found the support of Matthew, Sandro, San, Luca and Phil invaluable. The years have also seen support coming from Becky, Kristin, Cristina, Javi, Susana R, Aurélie, Colin, Natalia, Cesare, Susana F, Jose Luis, Nagore, Steve, Jen, Jean-Michel, Gabi and Claudio. Without friends, we are nothing.

Nomenclature

a	Speed of sound
b	Blade height
c	Chord
C_D	Drag coefficient
C_L	Lift coefficient
c_p	Specific heat at constant pressure
C_T	Tip clearance
c_v	Specific heat at constant volume
d	Rotor diameter
h	Blade height
H	Enthalpy
M	Mach number
N	Rotational speed
N_{corr}	Corrected rotational speed $\frac{N \cdot \pi \cdot d}{\sqrt{\gamma \cdot R \cdot T_{in}}}$
p	Static pressure
P	Stagnation pressure
P'_{out}	Ideal exit stagnation pressure $P_{in} \cdot \left(\frac{T_{out}}{T_{in}}\right)^{\frac{\gamma}{\gamma-1}}$
P_{loss}	Stagnation pressure loss $P'_{out} - P_{out}$
P_{loss}*	Modified pressure loss $\frac{\left(P_{out}' - P_{out}\right) \cdot \left(1 + \frac{\gamma - 1}{2} \cdot M_{in}^2\right)^{\gamma/\gamma-1}}{P_{in} \cdot M_{in}^2 \cdot \gamma}$
Q	Inlet flow function $\frac{W_{in}\sqrt{R \cdot T_{in}}}{A_{in} \cdot P_{in} \cdot \sqrt{\gamma}}$
Q_E	Crainic exit flow function $\frac{W_{in}\sqrt{R \cdot T_{in}}}{A_{in} \cdot P_{out} \cdot \sqrt{\gamma}}$

Q_{out}	Exit flow function $\dfrac{W_{out}\sqrt{R\cdot T_{out}}}{A_{out}\cdot P_{out}\cdot\sqrt{\gamma}}$
R	Gas constant
R_c	Pressure ratio
s	Blade width
t	Blade thickness
t	Static temperature
T	Stagnation temperature
W	Air mass flow rate
α	Air angle
α_m	Average air angle over blade: $\tan\alpha_m = \dfrac{\tan\alpha_1 + \tan\alpha_2}{2}$
β	Independent compressor characteristic parameter
β	Blade angle
γ	Ratio of specific heats
δ	Deviation $\alpha_2 - \beta_2$
Δ	Increase
ε	Error
ε	Deflection $\beta_2 - \beta_1$
η	Isentropic efficiency $\dfrac{\Delta H_{ideal}/T_{in}}{\Delta H/T_{in}}$
θ	Longwell loading parameter
μ	Viscosity
Π	Pressure ratio
ρ	Density
σ	Cascade solidity
τ	Torque
τ_{spec}	Specific torque $\dfrac{-\Delta H}{\sqrt{\gamma\cdot R\cdot T_{in}}\cdot N\cdot\pi\cdot d}$
φ	Flow coefficient Va/U
ψ	Work coefficient

Subscripts

in	At inlet
out	At outlet

Abbreviations

HP	High pressure
HPC	High pressure compressor
HPT	High pressure turbine
IP	Intermediate pressure
IPC	Intermediate pressure compressor
IPT	Intermediate pressure turbine
LP	Low pressure
LPC	Low pressure compressor
LPT	Low pressure turbine
RRAP	Rolls-Royce Aerothermal Performance (see Section 1.2.3)
UTC	University Technology Centre

1 Introduction

1.1 CRANFIELD UTC IN GAS TURBINE PERFORMANCE ENGINEERING

The second Rolls-Royce University Technology Centre at Cranfield was created in March 1998 to represent aspects of performance engineering. This project was started in October 1998 and forms the major part of Work Package 5 of the UTC, studying performance aspects of altitude relight and windmilling.

1.2 GAS TURBINE PERFORMANCE

1.2.1 Introduction

Performance modelling of a gas turbine engine takes place from its first conception to well into the engine's service life. Essentially, performance modelling refers to the analysis of the behaviour of the engine as a whole, rather than as individual components, primarily investigating the aerothermodynamics of the design. This analysis is then adapted to investigate overall performance measures and the effect on individual components. In the early stages of a new design, the preliminary sizing of components is performed by undertaking a design point analysis, in which sensible values of component efficiencies are used to produce a thermodynamic cycle which will fulfil the basic design criteria, usually relating to thrust and fuel consumption. Subsequent to the preliminary cycle design, further analysis takes place away from the design point conditions, in order to ensure that the operability of the engine, its emissions and noise profiles and various aspects of engine safety are satisfactory. Modelling may then continue beyond the design phase, in order to analyse in-service incidents and to schedule preventative maintenance.

1.2.2 Uses

Performance modelling is used in order to ensure the reliability of an engine design, to check that it has good fuel consumption, low emissions, good operability and low noise and to minimise the chance of producing an engine design which fails to meet specifications.

1.2.3 Overview of Off-Design Modelling Methods

Performance modelling is whole-engine simulation. As detailed computational models of the performance of the individual components of the engine would be very intensive on computing power, the components are treated as black boxes, otherwise known as *bricks*. These bricks generally contain experimental or computational maps of the component performances. These bricks are then joined together to build an overall model. The mass flow rate, pressure and temperature are conserved between the blocks and the appropriate relationships between the rotational speeds of the components are set where appropriate. If the simulation is steady-state, power or torque balances are made on the rotating components. The whole system is then iteratively solved.

1.2.4 RRAP and MARS

Many performance simulation packages are available for modelling gas turbines. Within Rolls-Royce, an internally-developed package is used, called Rolls-Royce Aerothermal Performance (RRAP). This is written in FORTRAN. Rolls-Royce are, however, moving to another, also internally-developed package, known as MARS and written in C++.

Both RRAP and MARS share many similar features. Particularly, they include a relatively robust multi-dimensional mathematical non-linear solver and a set of thermodynamic subroutines. These all form part of an overall gas turbine performance simulation tool, comprising the usual types of bricks to define the performance of individual components, linked together to form an overall system.

1.3 Windmilling and Relight

1.3.1 Introduction

This section introduces some of the background and terminology of the windmilling and relight of gas turbine engines. The various stages of the relight process are introduced in 1.3.2. The three main types of relight are then presented in 1.3.3. Finally, some of the uses of windmilling performance modelling are discussed in 1.3.4, with descriptions of some of the physical and numerical issues encountered.

1.3.2 Stages of Relight

1.3.2.1 Flame-Out

In certain adverse flight conditions, engines have been known to flame-out. That is, the flame inside the combustion chamber is extinguished and the engine then slows.

Engines are generally designed to make the chance of a flame-out small. However, they still occur, and so this must be considered a possibility in flight. To this end, techniques have been developed to relight engines in flight, effectively bump-starting them.

1.3.2.2 Spool-Down

Following a flame-out, the engine rapidly decelerates due to its loss of power. This transient is known as spool-down.

1.3.2.3 Windmilling

Although the combustion process is no longer active and therefore does not supply energy, the engine continues to rotate, driven by the dynamic head provided by the aircraft's forward motion. This rotation of the engine by the dynamic pressure is called windmilling.

1.3.2.4 Ignition

When the combustor inlet conditions are considered to be suitable, fuel may be injected into the combustion chamber and the igniters operated. In this way, ignition may be attempted.

1.3.2.5 Pull-Away

Following a successful ignition, the engine will normally accelerate, as in a normal start-up transient. This is known as pull-away.

1.3.3 Types of Relight

1.3.3.1 Introduction

There are three main types of relight: windmill relights, quick relights and starter-assist relights. The conditions of air speed and altitude at which each of these are used are shown in Figure 1.

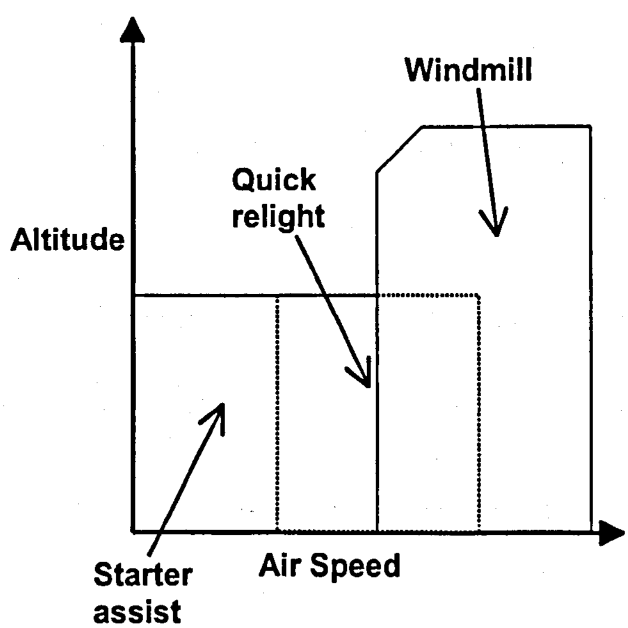

Figure 1 - The Three Main Types of Relight

1.3.3.2 Windmill Relight

At high flight speeds (generally above 250 knots) and at altitude, an aircraft may perform a windmill relight. This is the most common form of relight, where a flame-out has occurred during cruise. The aircraft therefore has sufficient altitude to permit a relight manoeuvre without the factor of time being a critical safety issue. The aircraft will generally descend, if necessary, to an altitude known to be within the relight envelope. If required, the engines are then purged of any residual fuel by windmilling them. Fuel is then injected and ignition and pull-away are attempted. The relight is considered successful if the engine reaches idle speed within a certain time, usually two minutes, from fuel injection.

1.3.3.3 Quick Windmill Relight

At take-off and climb conditions, an aircraft may perform a quick windmill relight. A relight under such conditions must be made very quickly, as the aircraft requires thrust and hydraulic power to continue to climb.

1.3.3.4 Starter-Assist Relight

At low flight speeds, an aircraft may perform a starter-assist relight, where the starter system is employed to provide additional power to the engine.

1.3.4 Importance of Improved Windmilling and Relighting Modelling

1.3.4.1 Introduction

Relight and power offtake availability at windmilling conditions are certification issues. Thus, any new engine design must satisfy a set of requirements in order for any engines to be sold. This section describes these and other aspects of windmilling and relighting that we wish to model. Then, phenomena associated with the physics of the gas turbine's operation and with the numerical methods used to perform the analyses are discussed.

1.3.4.2 Uses of Windmilling and Relight Modelling

WINDMILLING DRAG

The drag produced by a windmilling engine is an important issue for flight. This is for two reasons: the extra drag of the windmilling engine contributes greatly to the overall drag of the aircraft; and a situation in which one engine is windmilling will produce a yawing torque, which must then be corrected by the aircraft controls.

RELIGHT ENVELOPE

Figure 1 shows a generic relight envelope. This is generally a requirement for certifying a new engine.

At low flight speeds, the engine's starter may be used to rotate the engine's HP shaft, creating a pressure rise across the HP compressor and thus producing combustor conditions suitable for relight.

RELIGHT CONTROL LOGIC

A new engine must have its control system set up for good relight performance. This control logic must include:

- The purging of the engine to clear it of any unused fuel
- Determining at what point to start injecting fuel
- Determining the correct fuel flow rate to inject
- Ignition strategy
- Fuel schedule for pull-away for combustion efficiency and stability
- Variable geometry and bleed valve scheduling for compressor stability and good rates of acceleration
- Some account of the effect of transient heat soakage

These control logic factors can potentially be set up through modelling, using a synthesis code. However, in practice, much of the logic is determined by trial-and-error using an Altitude Test Facility (ATF).

MECHANICAL LOADS

During normal operation of the engine, when it produces thrust, the loads acting on the mechanical components of the engine, particularly the bearings, are in the opposite direction to the loads encountered during windmilling, where the engine produces drag. As the maximum reverse loads which the bearings will encounter are during windmilling, windmill operation becomes a design point for bearing design. It is therefore necessary to perform an analysis of engine behaviour to predict the loads which the mechanical components will experience.

Furthermore, during the relighting of the engine, hot spots will develop in the turbine. The synthesis model should ideally be capable of predicting whether the turbine inlet temperatures are within acceptable limits.

POWER OFFTAKES

As aircraft, particularly for military fighter applications, are requiring more and more use of hydraulic and electrical power for the operation of devices such as active aerodynamics, so there is an increasing requirement for the engines to provide this power, even during situations in which no engines are operational. On civil aircraft, a lack of hydraulic power can make extreme manoeuvres such as landing difficult; on fighter aircraft, the reliance on active aerodynamics can mean that simple flight is not possible.

1.3.4.3 Issues Encountered in Windmilling and Relight Modelling

IGNITION LOOP VARIATION

One stage of a successful relight is the ignition phase. Whether or not ignition is possible for a set of combustor inlet flow conditions is usually determined by an ignition loop. This ignition loop is normally created using a standard set of flow conditions and fuel supply conditions. However, these conditions may change for circumstances such as high altitude relight conditions. Therefore, work is necessary to model such effects as the variation of ignition loop due to fuel temperature, pressure changes or changes in the combustor metal temperature.

Combustor Efficiency and Stability

Subsequent to ignition, the flow through the engine will increase rapidly. This can then lead to conditions in which the combustor may flame out again after a successful ignition. Alternatively, the flow within the combustor may be such that the combustion efficiency is very low. This can lead to the acceleration of the engine slowing or stopping. A final associated problem is combustor rumble.

Heat Transfer

In any gas turbine transient modelling, an effect which must be taken into account is that of heat transfer. This is important for a number of reasons. Firstly, the bulk heat transfer in the main gas path leads to the situation whereby, in a compressor, the work done on the gas is not equal to the enthalpy supplied to it, or vice versa for a turbine. This is in contrast to steady-state running, where adiabatic processes may be reasonably assumed. Secondly, the expansion and contraction of the components leads to changes in their effective areas, their work-giving potential, their losses and their stabilities. Thirdly, the movement of the turbomachinery's discs leads to a change in internal seal clearances and hence a change in the internal flow behaviour of the engine. This can then lead to problems such as seal rub. In the processes of spool-down and of pull-away, the temperature changes which the components experience are greater than in normal transient operations and therefore these phenomena are of even greater importance for windmill and relight modelling.

Compressor Stall

During a quick relight, the engine experiences an extreme Bodie manoeuvre. The engine is initially running at normal operating conditions, in which the components are generally hot. Then, after a flame-out, the gas temperatures drop quickly and the turbomachinery components reject their heat. In the compressor systems, this results in the compressor blades reducing in height. However, as a relight is then quickly attempted, the casing has usually not begun to contract significantly. Therefore, tip clearances will be large. Then, when pull-away is attempted, the compressors, particularly the high pressure compressor, may stall.

Working Fluid Change

After a flame-out, the temperature of the gas stream reduces. As the specific heat of a gas is a function of the temperature, the components which normally operate with high temperature entry conditions will see a change in the specific heat of the fluid which is flowing through them.

In the turbines of an engine, the fluid which normally passes through is the products of combustion. However, while the engine is windmilling, it will pass only air. This then leads to the change of the gas constant of the turbine working fluid, although this change is generally small.

Characteristic Definition

During sub-idle operation, the components of the engine behave differently than at above-idle conditions. This can lead to problems with the definition of the operating point of the turbomachinery components. This is investigated in Section 3.2.

Sub-Idle Characteristic Generation

Most gas turbine performance models rely on the use of turbomachinery characteristics. These characteristics, or maps, are generally generated through a combination of design calculations, past experience with similar designs and tests on actual compressors. The most useful information usually comes from the last of these means, as this reflects the physical reality of a new compressor design. However, testing of compressors is very expensive and thus tests tend to be limited to the range of normal operation, usually above around 50% or 60% of design speed. During windmilling, far lower speeds than this are encountered and thus sub-idle turbomachinery characteristics must be generated by other means.

Instability of Operating Condition

During static operation of gas turbine engines, there is a power setting beyond which it is not possible to drop without reaching unstable operation. This is because the losses in the components are too great and is similar to the situation in a car engine encountered when trying to set the idle speed, where there is a certain fuel flow below which the engine will simply stall without any external load being applied. In gas turbine design, this effect is clearly important in the specification of a starter system, as the starter must be capable of accelerating the engine to the speed at which its operation is stable, known as the self-sustain speed. In most aeronautical engines, there is also a gearbox driven by the high pressure shaft of the engine. Because of the variation in the torque demanded by this gearbox to drive accessories, there is similarly a situation encountered in windmilling in which the HP shaft may have more than one possible operating point.

Fan Drag

Of importance for aircraft handling with a windmilling engine is the drag which the engine produces. In modern civil engines, the design point bypass ratio is very high, and increases dramatically when windmilling. Therefore, the fan usually provides the largest part of the windmilling drag. Thus, it is important to be able to determine the operating point of the fan accurately and the losses associated with this operating condition.

Core Windmilling Flow

The flow rate of air through the core of a windmilling engine largely determines its operating point. As the air flow experiences very little change in either temperature or pressure on its passage through the engine while windmilling, the density remains roughly constant. Therefore, the rate of flow of air is largely determined by the smallest flow area in the engine: the exit of the HP compressor.

Distorted Intake

In some military aircraft, the intake for the engines is bifurcated and thus the flow into the engine will be non-uniform. The effect which this has on the engine operation will be different for windmilling conditions than for above-idle operation. Furthermore, as windmilling could cause an aircraft to yaw, the intake of a normal turbofan engine could also be non-uniform.

Engine Variability

While all engines of a specific design may be expected to have the same performance as each other, in practice there can be significant variation between the performance of seemingly identical engines. Therefore, particularly for determining such things as relight envelopes and the unstable operation of an engine, account must be taken of this variability and the calculations performed for a worst-case engine.

Component Performance Degradation

During the lifetime of an engine, the performance of its components usually deteriorates. This then affects all aspects of the overall engine performance, including during windmilling and relight. Furthermore, one of the more well-known cases of flame-out on all engines was the result of an aircraft passing through the plume of a volcano. This led to the formation of coke on the engine components. Clearly, the issue of how an engine performs under such circumstances is important.

Windmilling at Extremes of Operation

After extreme failures, such as a fan-blade or an oil system failure, engines will normally be shut down. The drag produced by the engine is then of importance [31].

Reverse Windmilling

While considered outside the scope of this work, during ground starting, a tailwind may cause the engine to windmill backwards.

1.4 Scope of This Work

The field of modelling the performance of aircraft gas turbine engines during windmilling and relight is a very large subject area. Therefore, it was decided that this work should focus on some specific aspects of windmilling and relight.

It was decided to focus on cold-soaked, high altitude windmilling. Thus, this study does not investigate such phenomena as heat soakage in the turbomachinery or combustion efficiency, but nevertheless provides some ground work for further studies to expand the scope.

Furthermore, it was noted that the performance of the turbomachinery components was of greatest importance in determining the operation of an engine. Therefore, the work focuses primarily on the operation of compressors and turbines, with relatively little work on the other components of the engine. The means of representing the performance of the components and the means of predicting their performance during windmilling were studied.

Due to the influence of two-dimensional flow effects on fan performance during windmilling, it was decided not to study them, other than to regard them in the same way as compressors.

In addition to analyses of the performance of the turbomachinery, the means of solving the flow and torque balances which define the engine's operation in performance synthesis calculations are investigated.

During the work, all components are assumed to be free of faults or deterioration.

Within this scope, the main aim of the project was to improve modelling methods, specifically producing reliable models of turbomachinery performance and a robust solver mechanism, but also extending the research to other areas as required.

2 Literature Review

2.1 INTRODUCTION

In this section, a review of the literature relating to the problems of windmilling, relight and gas turbine performance modelling is presented. This has been arranged into investigations of windmilling and relight fundamentals, of gas turbine performance modelling, of the modelling of the turbomachinery components of an engine, of aspects of combustion processes on relight, of the other engine components, and of programming methods and numerical techniques for performance modelling and particularly for windmilling.

2.2 WINDMILLING AND RELIGHT

2.2.1. Windmilling

Curnock [20] describes the windmilling process for relighting an aero-engine, and some of the practical problems with doing so. Particularly, he remarks that the air flow tends to follow the path of least resistance. Thus, both spillage and bypass ratio increase when an engine starts to windmill. A streamline representation of this is shown in Figure 2.

He also notes that the outer parts of the fan blades frequently act as turbines, rather than as compressors, and that sometimes this applies to the whole fan. This turbine action then serves to accelerate the low-pressure shaft, on which the fan is located. However, the high-pressure compressor tends to act normally, in compressor mode.

The time to idle for relighting is not currently a legislative requirement for engines, but Curnock suggests that anything in excess of two minutes is undesirable.

Curnock also remarks on the smaller stall margin seen in a hot engine.

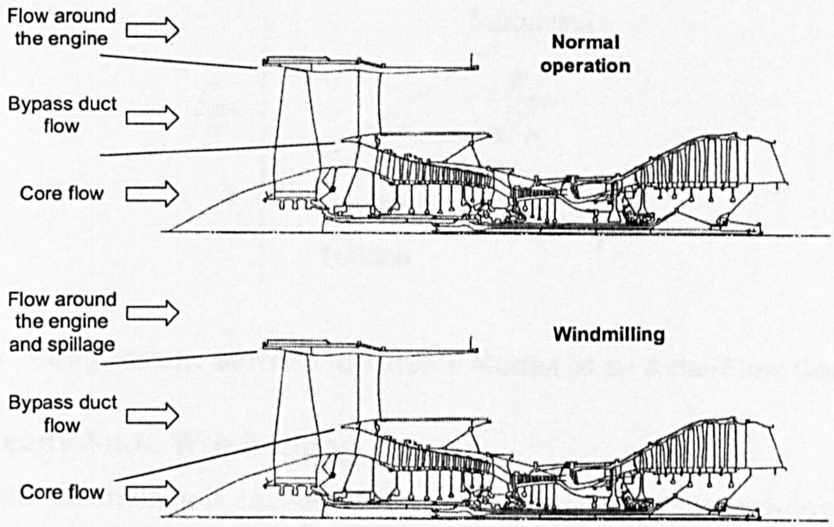

Figure 2 - Alteration of Spillage and Bypass Ratio Encountered when Windmilling

Jackson [31] provides a review of literature available within Rolls-Royce regarding windmilling and relight and their modelling, as well as describing many of the issues associated with them. He shows the way in which a compressor in windmilling can act as a compressor, stirrer or turbine through the diagram shown in Figure 3.

Walsh and Fletcher [82] also provide a good chapter on the windmilling phenomenon.

Owen et al. [61] highlight the problems associated with modelling the starting of a gas turbine in the windmilling regime:

- Little information exists in the low speed range of operation, particularly in respect of component maps.
- There is considerable variability between engines of the same model, especially with operating hours.
- Standard component maps do not include transient behaviour, such as heat soakage.

Clark and Green [15] write that the pressure spike associated with relight can cause compressor stall in a quick relight following a stall-related flame-out.

Braig et al. [8] provide a good paper on windmilling modelling. Here, they compare various engine configurations and describe a means of producing such an analysis.

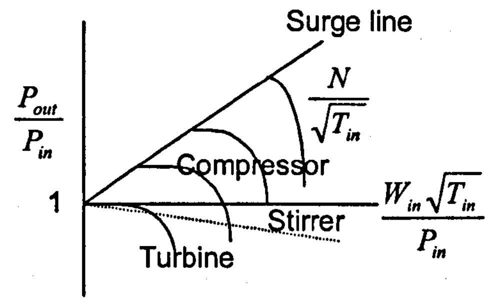

Figure 3 - Compressor, Stirrer and Turbine Modes of an Axial-Flow Compressor

2.2.2 Steady-State Windmilling

Steady-state windmilling is encountered when an engine has spooled-down and not relit. The modelling of this has been the subject of much work in the past.

Many reports are available from NACA from the 1950s [26],[74],[79],[80],[81],[83] on the topic of windmilling drag. This can be useful in producing the running line in the windmilling region. For fixed rotor windmilling, the internal drag defines the pressure ratio vs. mass flow relationship for the speed line $N/\sqrt{T} = 0$. This would then help substantially in the component characteristic extrapolation process. However, only one of these reports studies components individually [81]. Most of these studies [74],[80],[81],[83] were highly experimental, and thus specific to certain engines.

However, Hatch [26] suggests a solver routine for determining the maximum windmilling flow and speed. This is:

- Determine the choking flow for the last stator blade row of the compressor system.
- Use the total pressure loss (dependent on rotational speed) to predict the equivalent flow.
- Assume that the air angle leaving the last rotor is equal to the design value, in order to determine the rotational speed.

This routine is only valid for a single-spool engine.

The report also shows the general shape of a low speed compressor map.

Similar correlations can be made using ESDU 81009 [1]. This provides a method for predicting internal windmilling drag and airflow based on design specific thrust, Mach number, total pressure and total temperature. However, it only claims to be valid for a limited bypass ratio range. Being entirely steady-state, this only describes the working line of the engine, and no specifics as to where the drag is incurred.

A major problem with modelling steady-state windmilling is highlighted by Braig et al. [8]. In this paper, they state that it is possible to obtain two or more solutions for any given set of operating conditions, because of the influence of power off-takes. This is shown in Figure 4.

This condition of multiple solutions due to offtakes affects all shafts of the engine, although only the high-pressure shaft normally carries a gearbox, as shown in Figure 5. This is because a change in the speed of the HP shaft will inevitably produce a change in the mass flow rate, which alters the operating points of the other components also.

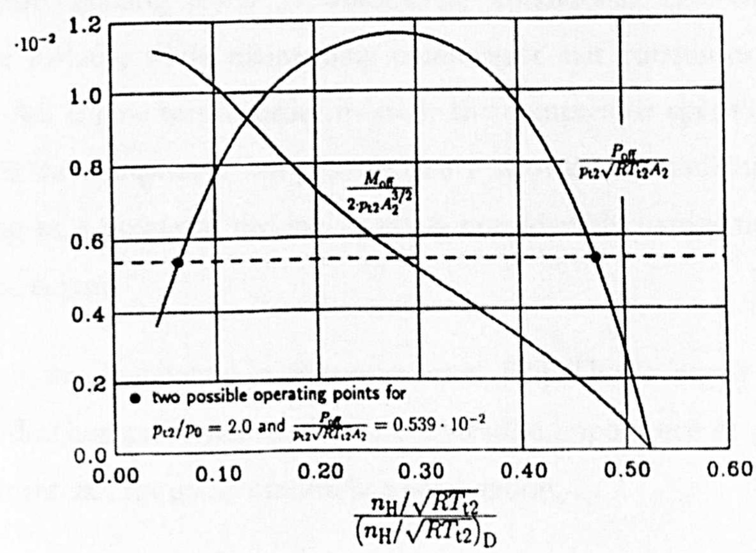

Figure 4 - Multiple Steady-State Solutions for a Given Set of Input Conditions Due to the Influence of Power Off-Takes [8]

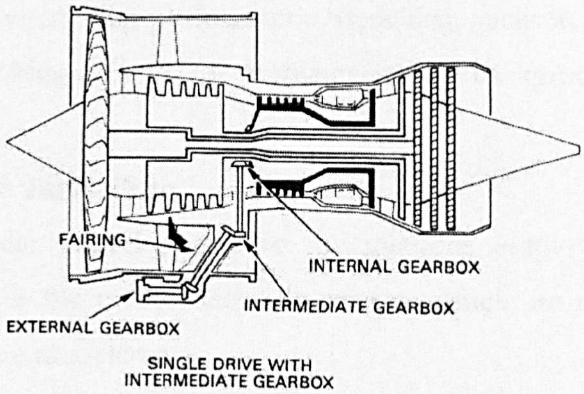

Figure 5 - A Typical Gearbox Configuration, on the High-Pressure Shaft of an Engine [68]

2.2.3 Windmilling Transients

In order to predict the windmilling relight process, the transient performance during windmilling must be modelled. Therefore, the transient characteristics of gas turbines in the low speed and windmilling region of operation must be investigated.

One of the NACA transient studies [80] looks at the deceleration of an engine under different braking loads in windmilling conditions. The researcher also removed the turbine, while maintaining compressor exit conditions the same as those in the full engine test in order to study the compressor specifically. He then mentions that the compressor can provide nearly all of the windmilling power. That is, it is acting as a powerful turbine, working considerably harder than the actual turbine of the engine.

Another study was conducted by Sosounov et al. [75]. This is largely a steady-state study of windmilling performance, but also notes the importance of a high turbine efficiency for the fan for good windmilling acceleration.

2.3 GAS TURBINE PERFORMANCE

2.3.1 General

A number of books provide good overviews of gas turbine operation in general. These include *The Jet Engine* [68] and *Gas Turbine Performance* [82].

On the subject of windmilling performance modelling, Jackson [31] has written a position paper, describing many of the problems and the background.

2.3.2 Steady-State Modelling

Curnock [21] provides a background to the methods employed to model gas turbines. He describes the mathematical constraints which are used to define the system for steady-state modelling:

- Flow continuity.
- Turbines and compressors on the same shaft operate at the same speed.
- Compressor work is equal to turbine work (minus any power offtakes or mechanical losses).
- Conservation of momentum.
- Component characteristics.

An investigation of the effects of power offtakes and bleed offtakes on steady-state operation of a gas turbine has been performed by Michaelides [55]. This shows the decrease in the surge margin of the compressors by increasing the load, and that good use of bleeds can increase surge margins.

Mechanical losses are investigated by Agrawal and Yunis [4]. Windage losses and other viscosity effects are modelled and expressions for the resultant torques are provided.

Walsh and Fletcher [82] also state the significance of bearing and windage losses at low speeds.

2.3.3 Transient Performance Modelling

MacIsaac and Saravanamuttoo [50] describe the principles behind transient performance modelling of gas turbines. An assumption of quasi-steady flow is used, permitting the use of steady-state performance maps to describe the components' operation. The difference in the torque generated by the turbine and absorbed by the compressor on each shaft of the engine is then calculated, and this is used to predict the acceleration, based on the rotational inertia of the components.

In transient performance modelling on a computer, there are two basic options available [62]: the mass flow method and the intercomponent volume method. The former assumes mass continuity at all times and points; the latter uses mass flow imbalances to calculate the pressure dynamics of the engine. Pilidis describes the methods of each and their strong and weak points. He also remarks that iterative convergence on the mass flow method is sometimes difficult.

Ibrahim [29] also provides a good comparison between the two methods. The requirement for very small time steps for the intercomponent volume method is noted, as is the need for more iteration with the mass flow continuity method. The intercomponent volume method was found to produce more accurate predictions through the first moments of a transient, while the mass flow continuity method was quicker for large transients.

2.4 TURBOMACHINERY MODELLING

2.4.1 Problems with Steady-State Component Characteristics

Most transient performance analyses assume that components have the same behaviour as in steady-state operation. However, a number of authors have noted highly transient effects which may modify the surge line and the characteristics themselves. Most of this work has been focused on the above-idle range of operation.

As mentioned above, Owen et al. [61] highlight the transient effects such as heat soakage as a problem with using steady-state component characteristics in order to model transient behaviour in gas turbines. They suggest [60] the modelling of the rotor and stator blades as complete items, rather than dividing them up, and also modelling their bases. They note that the engine's thermal state is important. Bleed control and inlet guide vane and stator vane variable operation can also affect the performance characteristics.

McLaughlin [54] also recommends that a transient simulation take into account engine structure temperatures and clearances.

Pilidis [62] expands upon this by including heat soakage and tip clearance calculations in the model used. This work builds upon that already produced by Lakshminarayana [41]. Pilidis' work showed that, under a standard (i.e. motoring) acceleration, the tip clearances initially reduce, as the blades expand under thermal and centrifugal stresses. The casing then expands, giving a larger clearance. Finally, the disc expands to reduce this again. These changes will have a significant effect on the ability of a compressor to produce a pressure rise. Thus, they affect the compressor map during a transient, particularly in respect of the surge line and the efficiency. Furthermore, the tip clearance changes are not the same for each stage of a single compressor. The former of these problems shows a limitation of steady-state characteristics, while the latter suggests that a technique such as stage stacking may be beneficial. However, MacCallum [49] describes a method of representing these problems without resorting to the use of individual stage characteristics. While this was shown to be reasonable in the work of Pilidis, it is possible that the effects during windmilling will be different. Pilidis and MacCallum [63] recommend the use of equivalent stages to model the compressor, rather than true stage characteristics. They also note that seal clearances vary during transients, and thus internal flows may alter.

The work of Ibrahim [29] expands on that of Pilidis and MacCallum, describing a heat transfer model in some detail. He also investigates the effect that tip clearances have on efficiency, the movement of seals and boundary layer effects. The thesis provides a valuable comparison between the modelling of transient effects with and without heat transfer. It is here noted that the heat transfer effects increase the time for the transient.

MacCallum [48] takes account of the heat transfer effects, and comments on the influence of endwall effects. The endwall boundary layers affect the operation of the compressor in two ways: restricting the flow area, and reducing the tip clearance losses. The boundary layers increase with pressure ratio and result in an efficiency loss. He also notes the importance of knowledge of the engine's past manoeuvres in determining the current performance characteristics.

Lakshminarayana [41] provides a flow efficiency correlation for the effects of tip clearances.

Bauerfeind [7] and Thomson [77] note three important transient effects:

- Heat exchange between gas and material.
- Packing lags due to the filling up of volumes.
- Transient combustion phenomena.

In the most part, the last of these effects is irrelevant to this study, combustion not being present until ignition occurs. Indeed, if the intercomponent volume method of transient performance prediction is used, the effect of packing lags is largely irrelevant also.

The heat exchange processes have two main influences:

- An absorption of heat into the components during an acceleration reduces compressor efficiencies and reduction of the surge margin through mismatching of stage characteristics
- Tip clearances alter as the blades, casing and discs expand.

Packing lag effects are only important over the initial stages of the acceleration, but arise because temperature and pressure changes are also accompanied by changes in density.

Bauerfeind [7] also notes that the inertia of the high-pressure spool has more influence than the inertia of the low-pressure spool during standard transients. However, this is not necessarily the case in windmilling, where the conditions at inlet to the low-pressure compressor or fan are varied, as opposed to the high-pressure turbine inlet conditions

Thomson [77] provides mathematical descriptions of these processes. This shows heat transfer rates, modifications to the component operating points and effects on efficiency.

2.5 Component Characteristic Extrapolation

Components' performances are rarely mapped in the low-speed regime, mainly because of the limitations of test rigs and instrumentation. Therefore, estimates must be made of their operation for windmilling performance calculations.

Joachim Kurzke [40] has done much work on the extrapolation of component maps to the low speed and low flow regions encountered in the study of windmilling.

Other methods for the mapping of components in the low speed, low flow region are presented by Riegler et al. [66], Agrawal and Yunis [4] and Yan and Mai [85].

Agrawal and Yunis [4] use a generic approach to predicting performance at starting speeds. The steady-state behaviour of the components is first defined, based on extrapolations. These points are then expanded towards surge and choke through the use of similar relationships.

Chappell and McLaughlin [12] base their extrapolations on the component characteristics. The point of zero speed, zero flow and unity pressure ratio is known, and the extrapolation can then follow from this. This approach is shown in Figure 6 for a compressor map.

Yan and Mai [85] suggest that extrapolations are more reliable if based on stage-stacking. This is a considerably more computer-intensive task, but many authors recommend its use.

These extrapolation methods are investigated in more detail in Section 3.3.

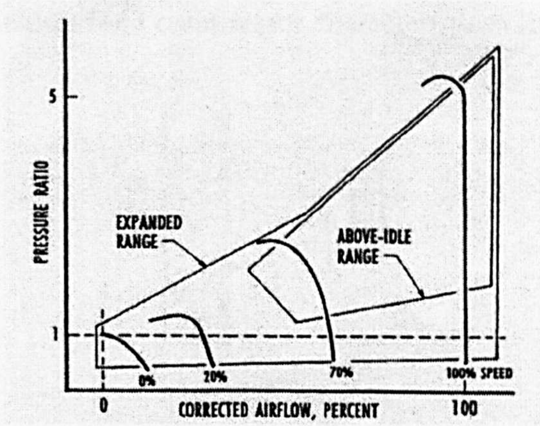

Figure 6 - Extrapolation of a Compressor Characteristic to the Low-Speed and Windmilling Region [12]

2.6 Component Characteristic Representation

There are two potential problems with the representation of component characteristics in certain forms. These are the variability of the performance depending on transient effects, as discussed above, and iterative problems from nonlinearities.

Like Yan and Mai [85], Seldner et al. [72] propose the use of stage stacking to represent the compressor characteristics. They state that this means that only functions of one variable need be considered. In addition, variable geometry and bleeds can be modelled more easily. However, they also note that its use is far more processor-intensive.

Schobeiri et al. [71] similarly model each stage of a turbine or a compressor individually, to account for heat transfer, bleeds and cooling air. These are separated by a small plenum to account for the packing lag effect.

The approach of individual stage characteristics is also proposed by Onions and Foss [59] to improve accuracy.

In order to aid iteration, the characteristics should be represented such that they are almost linear and have no discontinuities. This is not the case for standard component maps at low speeds.

The normal representation for a compressor characteristic is shown in Figure 7.

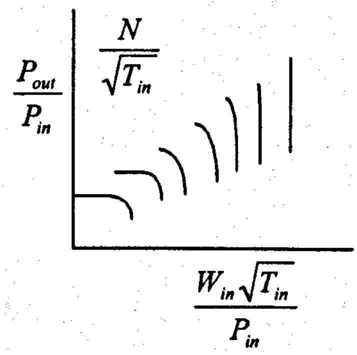

Figure 7 - Typical Representation of a Compressor Characteristic

At high speeds, the curves are almost vertical as they operate choked; at low speeds, a significant part of them is horizontal, with an almost constant pressure rise nor providing a large obstruction to the flow. In the low speed region, of interest to us, this means that a given rotational speed and pressure ratio is insufficient to describe the operating point of the compressor. Furthermore, as this horizontal part of the characteristic is approached, a Newton-Raphson style iteration could easily suggest pressure ratios which cannot be achieved at the given speed. Therefore, it would be sensible to investigate other means of representing the performance characteristics of a compressor for iterative solution purposes.

One method, which is commonly used, is that of Beta lines. By laying an arbitrary grid of lines over the compressor characteristic, nominally at right angles to the speed lines, the problem of multiple solutions to the same combination of pressure ratio and speed is avoided. This is shown in Figure 8.

Jackson [31] notes the way in which characteristics are frequently represented. This is reproduced in Table 1.

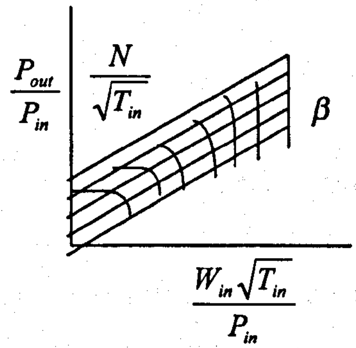

Figure 8 - Creation of Beta Lines to Aid Iterations

Table 1 - Representation of Component Characteristics in High and Low Speed Modelling [31]

	Independent variables		Dependent variables			
At high power						
Compressors	$\frac{N}{\sqrt{T_{in}}}$	β	$\frac{W_{in}\sqrt{T_{in}}}{P_{in}}$	$\frac{\Delta H}{T_{in}}$	η or $\frac{\Delta H_{ideal}}{T_{in}}$ or $\frac{P_{out}}{P_{in}}$	
Turbines	$\frac{N}{\sqrt{T_{in}}}$	$\frac{\Delta H}{T_{in}}$	$\frac{W_{in}\sqrt{T_{in}}}{P_{in}}$	η		
At low power						
Compressors	$\frac{N}{\sqrt{T_{in}}}$	β	$\frac{W_{in}\sqrt{T_{in}}}{P_{in}} \Big/ \frac{N}{\sqrt{T_{in}}}$	$\frac{\Delta H}{N^2}$		$\frac{\Delta H_{ideal}}{N^2}$
Turbines	$\frac{N}{\sqrt{T_{in}}}$	$\frac{\Delta H}{N^2}$	$\frac{W_{in}\sqrt{T_{in}}}{P_{in}} \Big/ \frac{N}{\sqrt{T_{in}}}$	$\frac{\Delta H_{ideal}}{N^2}$		

These representations are designed to avoid the iterative problems.

The accuracy of the performance model can also be affected by the choice of representation technique. In a number of papers and reports (including Mathioudakis [51] and [52]), in particular in AGARD [3], the use of an alternative working fluid is considered, in the study of humidity effects. In a windmilling gas turbine, the turbine components see such a change in working fluid, while the intermediate pressure and high pressure compressors also see a change in the specific heat capacity of the air. Therefore, these arguments are applicable to this study.

2.7 COMBUSTION

2.7.1 Relight

The performance model resultant from this project should aim to provide a simple model of combustor relight. This should take the form of a Boolean function based on pressure, temperature and mass flow at the combustor inlet to define whether ignition will occur. However, most published research appears to be focused on the combustion phenomena rather than on giving empirical correlations such as are required here.

A number of combustion texts describe some of the fundamental conditions required for ignition.

A typical relight envelope is described by Curnock [20] in Figure 9.

The top left corner of the windmill relight envelope is thought to be limited by poor combustion efficiency and fuel flow control problems. The right of the envelope shows the flight speed limitation caused by too high an airflow in the combustion chamber. This also produces too weak a mixture for combustion, as highlighted in a typical combustor stability diagram below. These must be produced for a range of combustor inlet pressures.

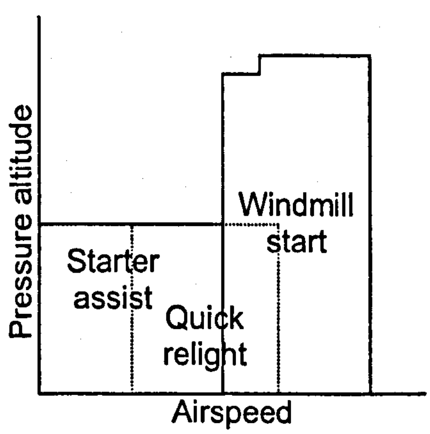

Figure 9 - A Typical Relight Envelope for an Aero Gas Turbine Engine

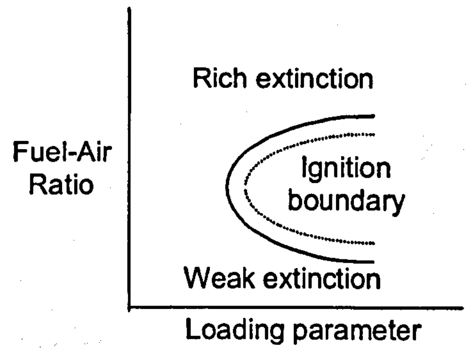

Figure 10 - A Typical Combustor Stability Diagram

A loading parameter such as that used in Figure 10 is defined by Longwell [47] (Eq. 1).

$$\textbf{Eq. 1} \quad \theta = \frac{P_4^{1.8} T_3 V_c}{W_4}$$

Kuo [37] describes the main criteria as temperature, time and turbulence. He then shows experimental results of how minimum ignition energy varies with flow speed, fuel droplet size and stoichiometry. These data also show a considerable random element to the ignition process. The mathematical ignition predictions presented in this book therefore require a study of the combustor flows in order to predict the flow speed at the point of ignition. The ignition energy, for the purposes of this study, can be assumed to be constant, and so the flow speed and temperature are required.

Lefebvre [44] writes of the representation of the ignition envelope as a three-dimensional chart relating fuel-air ratio, air mass flow rate and pressure for a given engine. The influence of inlet temperature is here incorporated in the mass flow rate and inlet pressure, as the correlation is specific to an engine. Such a combustor ignition loop is shown in Figure 11.

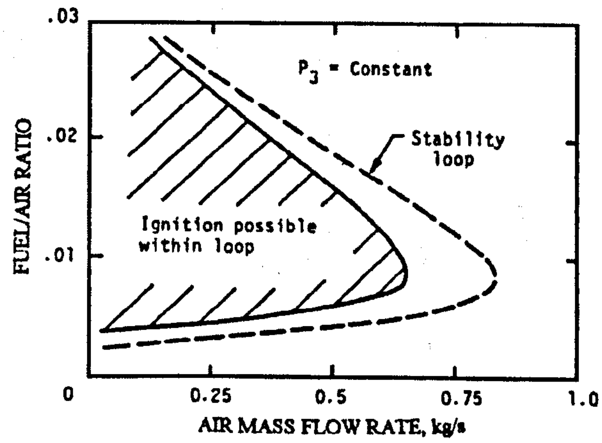

Figure 11 - Typical Combustor Ignition Loop [44]

A similar combustion stability and ignition loop is provided by Singh [73]. Singh also describes the three phases of ignition:

- Formation of a flame kernel.
- Propagation of the flame to all parts of the primary zone.
- Light-around to all other segments in tubular or tubo-annular combustors.

It is noted that conditions can be such that the first stage can occur, but not the second.

Lefebvre [44] also notes that the spark energy and duration are of importance.

Ballal and Lefebvre's paper [6] provides relationships for finding the minimum ignition energy, for a range of droplet sizes, combustor flow velocities, temperatures, pressures and turbulence levels. However, such a correlation requires knowledge of the velocity at the point of ignition, of the turbulence at that point, and the droplet sizes resultant from these conditions. Thus, it is of limited direct use in this model.

The title of Lefebvre's report [45] suggests that it is probably an excellent starting point for building the combustion part of the model. However, the report itself turns out to be very theoretical. This means that the drawing of an empirical correlation would be difficult, requiring a detailed study. Rather, an entirely empirical method is needed, either from rig tests or from previous computational work. Nevertheless, the correlation resultant from this report is of use. This is presented in Eq. 2.

$$\textbf{Eq. 2} \quad I = f\left[\frac{E^{0.25}P_2^{\ 0.5}T_2^{\ 2.5}}{U_{ref}\left(\dfrac{\Delta P_{ft}}{q_{ref}}\right)^{0.5}}\right]$$

where: I = ignition performance

E = effective thermal component of total ignition energy

P_2 = inlet air pressure

T_2 = inlet air temperature

U_{ref} = chamber reference velocity

ΔP_{ft} = flame-tube pressure drop

q_{ref} = dynamic head based on inlet pressure and temperature and U_{ref}

Therefore, a similar correlation would be expected from experimental and computational data, and thus the above relationship may be useful for interpreting such data.

2.7.2 Combustion Efficiency and Stability

Lefebvre [42],[43],[44] describes the use of stability loops to define whether the combustor can achieve reliable combustion at a given set of operating conditions. Such a stability loop is shown in Figure 11. He also notes that the stability of the combustor is affected by the combustor pressure, as shown in Figure 12.

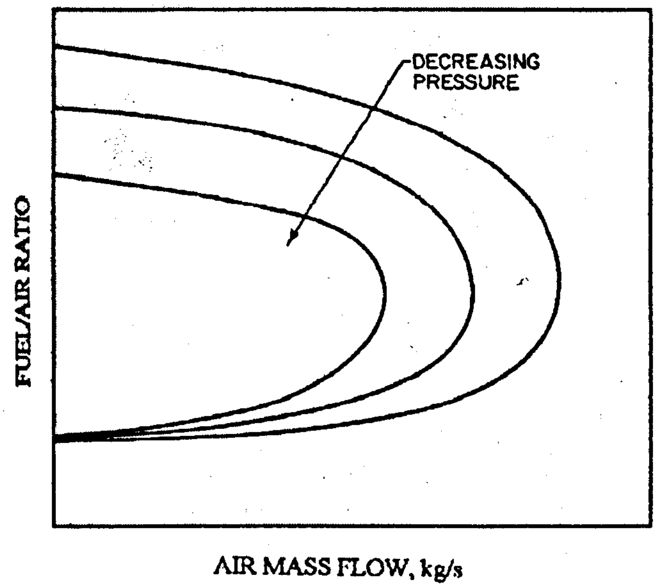

Figure 12 - Effect of Pressure on Combustor Stability Loop [44]

Lefebvre [44] notes that the combustion efficiency is influenced by the evaporation rate, mixing rate and reaction rate. This is investigated in more detail in [43].

Ibrahim [30] applies the work of Lefebvre to an engine, with a view to studying the combustion efficiency during the pull-away stage of relight.

2.8 Other Components

Much information on modelling the off-design performance of the other components of a gas turbine engine is available in Walsh and Fletcher [82]. However, little is available which is specific to sub-idle modelling, although most components are not expected to behave significantly differently during windmilling than at above-idle conditions.

A good report was written within BMW Rolls-Royce (now Rolls-Royce Deutschland) by Oberursel [58] on the power requirement of the accessory systems during windmilling.

2.9 Programming and Numerical Considerations

2.9.1 Programming Considerations

Schobeiri et al. [71] have created a code which is highly modular in design. Here, components are coupled at run-time, rather than having to be hard-coded. This saves considerable effort in the simulation of alternative engine configurations.

Curlett and Felder [19] make the observation that batch and graphical processing in the same package are desirable. Using such an approach, it would be possible to map a relight envelope, or to study the details of a particular transient.

The thesis produced by Adamopoulos [2] describes the workings of a code very similar to that produced by Dr. Yin at Cranfield University. This also includes a routine for a mixer, which may be useful in modelling such engines in windmilling conditions.

Saravanamuttoo and MacIsaac [70] remark that models should be suitable for any engine and any conditions. Thus, while the model to be developed in this project is essentially designed for windmill relight, it should also be suitable, perhaps with a little modification, for pull-away studies, when the main control handle will be fuel flow. They also note that the simulation program should require a minimum of effort in configuring to alternative engines, and that results should be presented in an easily understandable form.

Curlett and Felder [19] describe an alternative method for calculating the performance of a gas turbine engine. This is based on object-oriented programming. One of the advantages of such an architecture, which is cited, is the ability to zoom in on parts of the model. This then allows the use of coarse routines to obtain an approximate solution, and detailed routines to refine the solution obtained. For instance, standard component mapping techniques could be used in the first instance, moving to two- or three-dimensional analyses to refine. These techniques can be implemented in any programming language, but one, such as C++, which is designed for object-oriented programming, will be somewhat easier. Particularly, it is noted that the use of common block architectures, such as are employed in the current codes at Cranfield, is best avoided.

2.9.2 Iterative Convergence

Problems have been encountered in the modelling of gas turbines, particularly at low speeds, in the iterative process. Therefore, the convergence of the iterative process needs to be studied further.

McLaughlin [53] talks about the solver algorithms in some detail. Firstly, he describes the physical phenomena which define the engine's behaviour:

- Each stream uses the inlet flow as an iteration variable and the exit state boundary condition as a convergence criterion.
- Each shaft must have its speed as an iteration variable and a sum of zero torque as an exit boundary condition (for steady-state behaviour).
- Each rotating component uses pressure ratio (or some related function such as Beta) as an iteration variable, and flow continuity at inlet as a convergence criterion.

He states that the flow continuity requirement can cause iterative convergence problems at low speeds. This is because of the problems associated with the highly linear and horizontal speed lines at low rotational speed in compressors, as described above. The near vertical lines seen in turbine maps at low speeds create similar problems.

Unusually, McLaughlin promotes the use of static pressures, temperatures and densities, as opposed to their total equivalents. He claims that this causes quicker and more reliable iterative convergence at low speeds.

He also recommends the use of the Newton-Raphson method of nonlinear solution, augmented by a matrix based on Broyden's method [10].

Ganji et al. [23] use a multi-dimensional Newton-Raphson method, attempting to reduce all the matching errors simultaneously. In transient analysis, all the relationships, using an intercomponent volume method, are reduced to ordinary differential equations and solved thus.

Owen et al. [60] use an unusual iteration in that they vary P_6 rather than P_5 as described in Figure 13. They claim that this substantially improves convergence, as P_6 does not vary greatly.

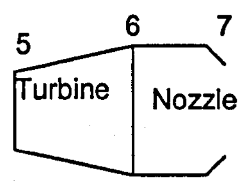

Figure 13 - Station Numbers Used for Owen et al. [60]

2.10 SUMMARY

A significant amount of public-domain literature has been found covering most areas of interest in the performance modelling of windmilling and relight. However, few studies have been conducted which study all aspects of the problem together. Information has been found covering the fundamentals of gas turbine performance modelling, windmilling and relight, turbomachinery characteristics for windmilling, combustion aspects of relight and windmilling performance modelling.

3 Turbomachinery Characteristics

3.1 INTRODUCTION

Two major issues are encountered by the performance engineer in modelling the sub-idle operation of the turbomachinery components of a gas turbine engine. These are the methods of representing the components within the overall performance algorithm and the generation of reliable data. These two issues are explored in the following sections.

3.2 TURBOMACHINERY REPRESENTATION

3.2.1 Introduction

In performance modelling of gas turbine engines, it is customary to represent the performance of the individual components of the engine through characteristics or maps. These are tabulated or graphical data which describe the operating behaviour of the component.

In these characteristics, the operating point of the component is described using a small number of variables (for instance pressure ratio and non-dimensional rotational speed). The remaining parameters of the map (for instance non-dimensional flow rate and efficiency) may then be used to determine the complete relationship between inlet and outlet conditions.

The choice of which variables to use for describing the performances of the components is important, as inappropriate choices may result in an ambiguous operating point, predictions of performance which are not correct, or the requirement of using maps of more dimensions than ideally required. Therefore, throughout the next section, the analysis of a variety of representation techniques is presented, along with recommendations for a representation scheme for use in windmilling modelling. The analysis is first conducted for compressors and then for turbines.

3.2.2 Non-Dimensional Similarity

The use of dimensionless or quasidimensionless groups to describe the operational characteristics of turbomachinery is common. As discussed in Walsh and Fletcher [82] and a number of other sources, the principle of maintaining Mach number similarity provides the foundation for turbomachinery characteristics. Neglecting second order effects, it can be demonstrated that, maintaining constant rotor and axial Mach numbers, the incidence of the air entering the first rotor is constant. This then mean that, to a first order approximation, the flow turning is constant. This leads to a constant angle of incidence for entry to the stator, and so it continues throughout the component. Resultant from this and from Euler's equation, we find that the enthalpy change is proportional to the square of the rotor speed. As the rotor speed is the Mach number multiplied by the sonic velocity and the Mach number has been held constant, it follows that the enthalpy change is proportional to the square of the sonic velocity. The losses in the turbomachinery do not follow as neatly as the enthalpy changes. However, it is often regarded as reasonable to assume that the pressure loss is proportional to the dynamic head of the airstream. As the dynamic head is, for a fixed Mach number, proportional only to the inlet pressure, the losses within the compressor can be mapped according to the Mach number principle also.

3.2.3 Compressor Representation

The compressors of an engine are probably the most important of the turbomachinery components in determining windmilling performance. The fan is the largest element of windmilling drag, and the HP compressor is the device which largely dictates the mass flow rate of air through the engine during windmilling. Therefore, it is imperative that the compressor characteristics be represented reasonably accurately in a windmilling performance simulation.

The choice of how to represent the performance of the fans and compressors of a gas turbine engine in a whole-engine performance model can be critical to the number of iterations required by the solver or indeed whether the system can be solved.

This section looks at the way in which compressor characteristics can be represented in a gas turbine performance simulation program. A paper was published on this topic by Jones et al. [33] at Turbo Expo 2001 as part of this doctorate. The issue is addressed in more detail in this section.

3.2.3.1 Variables for Compressor Operating Point Definition

Under many steady-state simulations, the speed of rotation of the shaft is known, as it is often used as the handle for calculations. Similarly, it is known for transient calculations as the speed is calculated from the torque imbalance on the shaft, its rotational inertia and the time step. As the inlet stagnation temperature for the LP compressor is equal to the ambient stagnation temperature, so the corrected speed, N_{corr}, is known. Similarly, for subsequent compressors, the inlet stagnation temperature is known from the calculations for the preceding compressors. Therefore, the corrected speed parameter is invariably used to define the compressor's operating point.

Using non-dimensional or quasi-non-dimensional representations of the operation of compressors, it is possible to define the operating point using just two variables, if the effects of Reynolds number, working fluid changes, variable geometry, bleeds, flow distortions, volume packing and heat transfer are neglected [82].

The further variables of variable geometry and bleed flows can be modelled by the use of further tables of properties. Further research into the effects of bleed flows, variable geometry, flow distortions and heat transfer is ongoing at Cranfield in the Rolls-Royce University Technology Centre in Gas Turbine Performance Engineering.

REQUIREMENT FOR NO COLLINEARITY

In choosing two variables to define the operating point, it is necessary that the variables must not be collinear and that each pair of variables should produce a unique operating point.

One of the most simple and intuitive methods of describing the compressor characteristic is the plotting of pressure ratio against flow and speed, as shown in Figure 14. While this defines the operating point at above-idle speeds, stating only pressure ratio and speed does not work for low speeds, as the two variables become collinear (horizontal on the chart). Thus, this representation cannot be used for performance calculations in the sub-idle region.

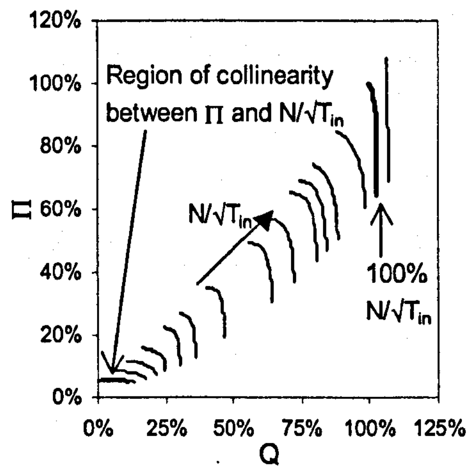

Figure 14 - Compressor characteristic definition using pressure ratio and speed

One way of overcoming the above problem is to use an arbitrary "Beta" variable to define the characteristics. This variable can be described in various ways, one of which [40] is to choose Beta=1 to be the surge line and lower values of Beta as roughly parallel to this, down to a minimum of zero, as shown in Figure 15. A characteristic for determining flow from speed and Beta is shown in Figure 16.

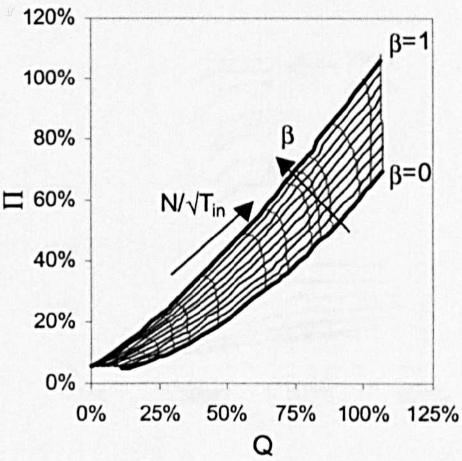

Figure 15 - Definition of Beta lines

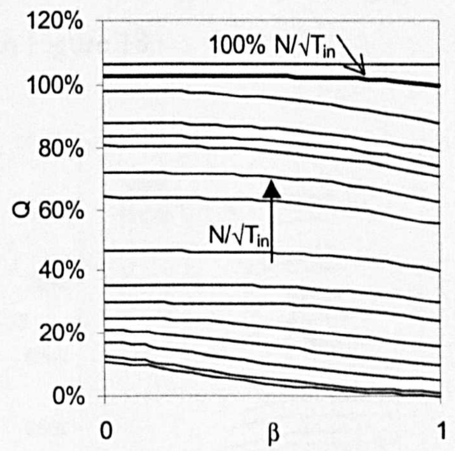

Figure 16 - Characteristic for defining operating point using Beta and speed

Such a representation using Beta and speed as the defining variables clearly works over the entire operating range of the compressor, with no collinearity between the two variables.

There also exist a number of other representations which satisfy the requirements for no collinearity between the defining variables.

Crainic et al. [18] suggest the use of the flow function Q_E. This gives good definition of the operating point, as shown in Figure 17.

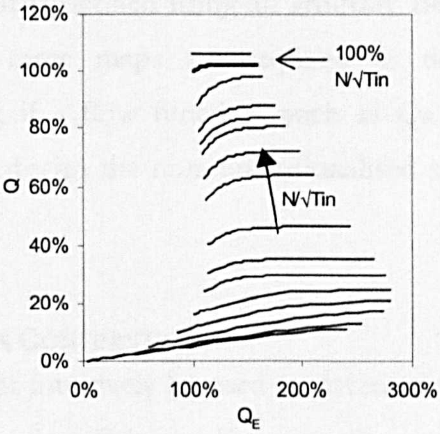

Figure 17 - Characteristic for defining the operating point using Q_E and speed

Another potential variable for defining the operating point is provided by the exit flow function as shown in Figure 18.

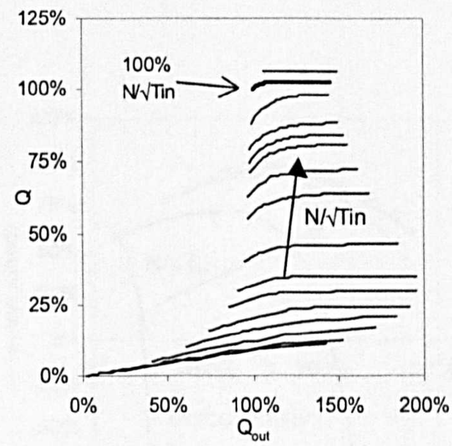

Figure 18 - Characteristic for defining the operating point using Q_{out} and speed

3.2.3.2 Characteristics for Determining Gas Properties

NUMBER OF MAPS REQUIRED

Once the operating point of the compressor has been defined, it is then necessary to determine the exit conditions of the air stream.

Together, the maps must define by some means the change in stagnation pressure and stagnation temperature, as well as the flow rate through the compressor and the speed at which it is rotating.

When the operating point is defined using an arbitrary Beta variable and the non-dimensionalised speed, three maps are required to describe the compressor conditions. Alternatively, if a flow function, such as Q_E or Q_{out}, or some other physical parameter is used with the non-dimensionalised speed, only two maps are required.

REQUIREMENT FOR CONTINUITY

One variable which might intuitively be used to determine the exit conditions of a compressor is the isentropic efficiency. Knowing the pressure ratio at which the engine is operating and knowing its efficiency, the exit temperature can be determined. However, efficiency reaches a discontinuity (Figure 19) when the pressure rise is zero [66], as is sometimes the case at low speeds. This discontinuity means that the exit temperature from the compressor cannot be determined at such conditions.

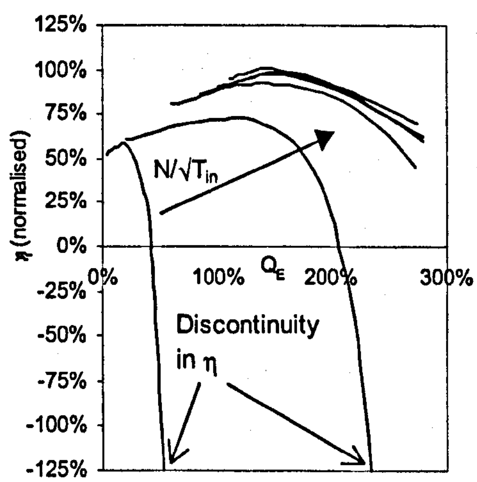

Figure 19 - Characteristic for determining the efficiency from Q_E and speed

During normal operation, the compressor behaves as was intended, with a temperature rise and pressure rise across its length. However, as the speed drops, it may enter two alternative modes of operation. At low flows, a compressor can behave as a stirrer, with a temperature rise but a drop in pressure due to losses. At higher flows, the compressor can act as a turbine, with a stagnation temperature drop (and hence an accelerating torque on the shaft) and a pressure drop. Both these types of behaviour can be seen in windmilling engines. It is in moving from the stirrer to the turbine modes of operation that the efficiency reaches a discontinuity, with lower flows producing negative values of efficiency and higher flows producing positive values.

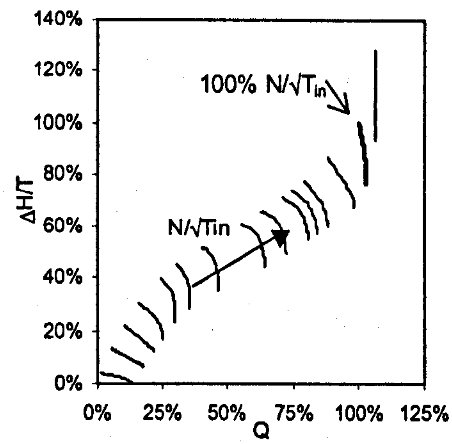

Figure 20 - Characteristic for determining the enthalpy rise from the speed and inlet flow

Instead of using efficiency directly to define the relationship between work input and pressure ratio, it is therefore necessary to define each independently of the other. It thus follows that the temperature ratio, temperature rise or enthalpy rise should be used. A plot of enthalpy rise is shown in Figure 20.

Characteristics described using enthalpy rise and either ideal enthalpy rise or pressure ratio remain finite over the entire operating range of the compressor, and thus the use of efficiency is not necessary.

Another alternative way of describing the inefficiencies in the compressor is the use of pressure loss (Figure 22). Describing the pressure drop through the turbomachinery, this analogy is also close to the physical reality.

DEFINITION FOR ZERO ROTATIONAL SPEED

The enthalpy rise for a locked rotor is zero for adiabatic conditions. Nevertheless, the compressor experiences a torque when there is flow. However, it is not possible to derive the torque from the enthalpy rise at zero speed [66]. As this torque is important for determining the starting acceleration of the LP and IP shafts of the engine, it is necessary to define it. Therefore, a better representation than the use of an enthalpy rise is to use specific torque, also known as corrected torque, as calculated in Eq. 3. The specific torque, plotted against Beta and speed in Figure 21, remains finite over the entire operating range and describes the characteristic at zero rotational speed.

$$\text{Eq. 3} \quad \tau_{spec} = \frac{-\Delta H \cdot}{\sqrt{\gamma \cdot R \cdot T_{in}} \cdot N \cdot \pi \cdot d} = \frac{-\Delta H / \gamma \cdot R \cdot T_{in}}{N \cdot \pi \cdot d / \sqrt{\gamma \cdot R \cdot T_{in}}}$$

3.2.3.3 General Requirements for Compressor Characteristics

LINEAR INDEPENDENCE

All the parameters used in the maps must be linearly independent. Failure to satisfy this requirement will produce a set of relationships which cannot be solved.

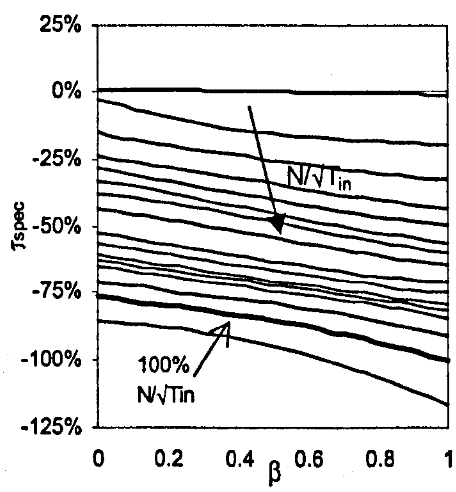

Figure 21 - Characteristic for determining the specific torque from Beta and speed

CHARACTERISTIC STORAGE AND INTERPOLATION

As characteristics are normally stored in look-up tables, the number of look-ups should be minimised, as this is generally expensive in terms of computation time. While the use of Beta has very little adverse impact on the memory required to store a characteristic, requiring only one extra column of data, the necessity of using three look-up tables instead of two will affect the computational speed.

Storing the characteristics in tabular form requires the use of interpolation to determine the conditions at operational points other than those tabulated. Here, it is preferable to have smooth functions as the order of the interpolation can potentially be reduced and the reliability of the interpolation improved.

It should also be noted that the use of Beta requires interpolation of three tables rather than two in order to define the speed line. Then, three interpolations are needed to determine the operating conditions as opposed to the two required when using a physical parameter for operating point definition.

An advantage of using Beta in this respect is that it produces a cuboidal matrix, with Beta ranging from zero to unity and speed ranging from zero to the maximum operating speed. This is in contrast to characteristics defined using Q_E, where the speed lines for lower speeds cover a larger range of Q_E than those for higher speeds. This factor makes interpolation slightly simpler.

One type of variable which is sometimes used for characteristics is pressure loss. This allows the concept of efficiency to be used without encountering the problems associated with efficiency itself. However, this parameter experiences a minimum, at roughly the design incidence, and thus interpolation may produce errors here unless a number of points are located near to this minimum. This should be relatively easy as the compressor would normally be expected to be operating at close to its design incidence, although this is not always true for low power settings. A plot of pressure loss against Q_E and speed is shown in Figure 22. This is similar to the technique used by Converse and Giffen [17] for generation of compressor characteristics.

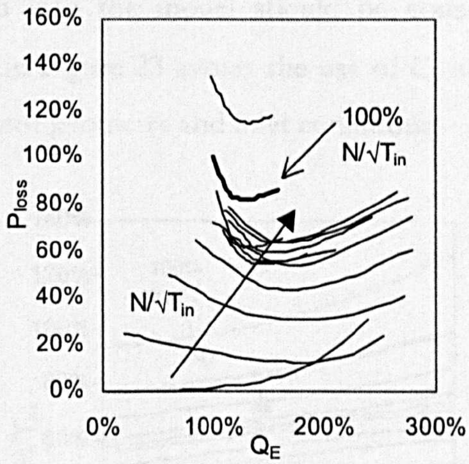

Figure 22 - Characteristic for determining the pressure loss from Q_E and speed

3.2.3.4 Incorporation of Other Effects [82]

TRANSIENT HEAT SOAKAGE

In transient calculations, the effect of heat soakage is of great significance. Much work on the subject has been carried out where transient modelling has been refined to include these effects on the behaviour of the gas and the performance of the components [48].

INTERSTAGE BLEEDS

As the opening and closing of bleed valves affects the relationship between inlet flow and exit flow, it is preferable to use Q_E rather than Q_{out} for characteristic representation.

REYNOLDS NUMBER EFFECTS

At low speeds and flows, the effect of Reynolds number may become significant. These variations mainly impact on the viscous losses through the compressor and thus may be incorporated through a modification to the pressure loss parameter.

INLET TEMPERATURE

As the inlet temperature changes, so too do the values of C_p and γ. While this effect is of less significance in compressors than in turbines, as the temperature variation is lower, its incorporation into the model should be considered. The use of an enthalpy rise, as shown in Figure 23 avoids the use of C_P and γ, essentially being a function of the compressor geometry and inlet conditions.

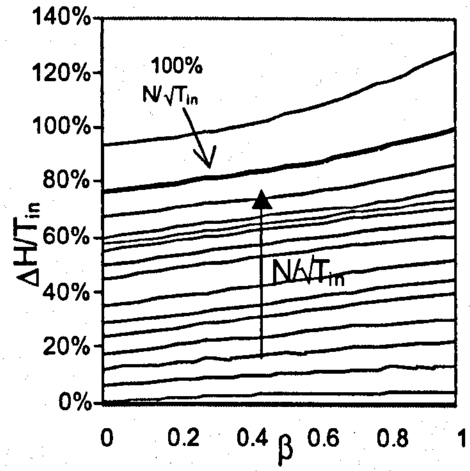

Figure 23 - Characteristic for determining the enthalpy rise for a given β and speed

The use of Q_E implicitly includes the pressure ratio. Thus, a change in the pressure generating capability of the compressor changes the operating point as defined by Q_E, as shown in Eq. 4. This substantially strengthens the case for using a Beta, defined using non-dimensionalised enthalpy rise and rotational speed.

Eq. 4 $$Q_E = \frac{W_{in} \cdot \sqrt{R \cdot T_{in}}}{A_{in} \cdot P_{out} \cdot \sqrt{\gamma}} = \frac{W_{in} \cdot \sqrt{R \cdot T_{in}}}{A_{in} \cdot P_{in} \cdot \sqrt{\gamma}} \cdot \frac{P_{in}}{P_{out}}$$

The use of a pressure loss to describe the inefficiency of the compressor will also be affected by a change in inlet temperature.

A significant proportion of the pressure loss may be expected to be proportional to density and the square of flow speed for small disturbances from an operating point [27]. Using this assumption, it is demonstrated below that the modified pressure loss parameter described in Eq. 5 would be expected to produce characteristics largely independent of inlet temperature. A characteristic produced using this parameter is shown in Figure 24.

$$P_{Loss} = P_{out}' - P_{out} \propto \rho \cdot V_a^2$$
$$\Rightarrow \frac{P_{Loss}}{\rho \cdot V_a^2} = const$$
$$p = \rho \cdot R \cdot t \Rightarrow \rho = \frac{p}{R \cdot t}$$
$$V_a = M \cdot a$$
$$a = \sqrt{\gamma \cdot R \cdot t}$$
$$\Rightarrow V_a = M \cdot \sqrt{\gamma \cdot R \cdot t}$$
$$\Rightarrow \rho \cdot V_a^2 = \frac{p}{R \cdot t} \cdot M^2 \cdot \gamma \cdot R \cdot t$$
$$\Rightarrow \rho \cdot V_a^2 = p \cdot M^2 \cdot \gamma$$
$$\frac{P}{p} = \left(1 + \frac{\gamma - 1}{2} \cdot M^2\right)^{\frac{\gamma}{\gamma-1}}$$
$$\Rightarrow p = \frac{P}{\left(1 + \frac{\gamma - 1}{2} \cdot M^2\right)^{\frac{\gamma}{\gamma-1}}}$$
$$P_{Loss} = P_{out}' - P_{out}$$

Eq. 5 $$P_{Loss}{}^* = \frac{\left(P_{out}' - P_{out}\right) \cdot \left(1 + \frac{\gamma - 1}{2} \cdot M_{in}{}^2\right)^{\gamma/\gamma-1}}{P_{in} \cdot M_{in}{}^2 \cdot \gamma}$$

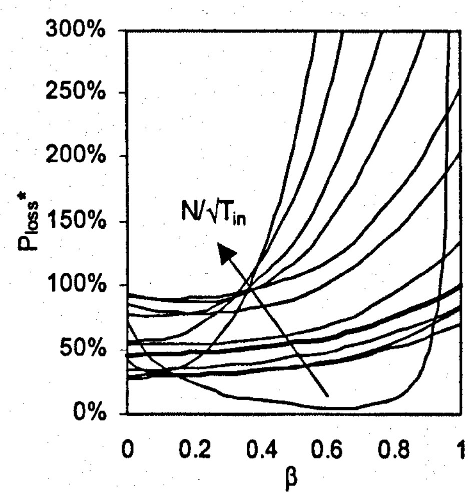

Figure 24 - Characteristic for determining the modified pressure loss from β and corrected speed

Similar effects to those seen with a variation in inlet temperature are also seen where the humidity or gas composition varies, changing the specific heats.

VOLUME PACKING

During transients, the flow into the compressor may not be equal to the flow out of it due to the effect of density changes. Thus, the use of a true exit flow function, Q_{out}, is not advised. Q_E is not affected in this way.

3.2.3.5 Conclusions

MATCHING PARAMETERS

Corrected speed is invariably used as one of the two parameters necessary to define the operating point of the compressor. A further variable is then required. The main options for this variable are described in Table 2.

Table 2 - Matching Parameters for Compressor Characteristics

Matching parameter	Comment
Pressure ratio	Multiple solutions at low speeds and flows. Inlet temperature dependent.
Outlet flow	Dependent on bleeds and volume packing. Incorporates pressure ratio and thus not independent of inlet temperature.
Q_E	Incorporates pressure ratio and thus not independent of inlet temperature.
β	Uniquely defines operating point. Independent of inlet temperature. Requires three look-up tables rather than two.

INDEPENDENT VARIABLES

When β is used as the matching parameter, three independent variables are required to determine the flow conditions. One of these is invariably the inlet flow function, Q_{in}. Table 3 looks at the remaining possible variables.

Table 3 - Independent Variables for Compressor Characteristic Representation

Independent variables	Comments
Π, η	Pressure ratio dependent on inlet temperature. Efficiency does not remain finite and is not independent of inlet temperature.
$\frac{\Delta H}{T_{in}}, \frac{\Delta H'}{T_{in}}$	Zero-speed torque not defined. Ideal enthalpy rise not independent of inlet temperature.
τ_{spec}, P_{loss}	Zero-speed torque defined. Pressure loss not independent of inlet temperature.
τ_{spec}, P_{loss}^*	Zero-speed torque defined. Modified pressure loss largely independent of inlet temperature.

RECOMMENDATIONS

When modelling the performance of a gas turbine engine over the entire operating range, the use of pressure ratio for operating point definition and the use of efficiency for calculation of exit conditions presents some problems. These representations do not satisfy the requirements of defining a unique operating point and of remaining finite and continuous.

The use of Beta for defining the operating point of the engine has an advantage over Q_E and Q_{out} in that it is independent of inlet temperature.

Specific torque provides good definition of the stagnation temperature at outlet from the compressor over the entire operating range, while also defining the torque experienced by the shaft of a locked rotor. Specific torque is advantageous over the enthalpy rise as this does not define the operating conditions fully at zero speed.

The modified pressure loss is suitable for completing the calculation of the compressor outflow conditions, although care should be taken to define it precisely at near to the design incidence.

3.2.4 Turbine Representation

As with compressors, problems are encountered when trying to represent turbines in a gas turbine performance package at the low speeds encountered during windmilling. Therefore, techniques for representing the turbine performance have been investigated. Since the same arguments hold, in general, for turbines as for compressors, many of the arguments are brief.

3.2.4.1 Variables for Turbine Operating Point Definition

REQUIREMENT FOR NO COLLINEARITY

In order to uniquely define the operating point of a turbine, there must be no collinearity of the variables. However, at choking flows, collinearity occurs between non-dimensional flow and non-dimensional speed. At low speeds, it is encountered if the variables of expansion ratio, or ideal enthalpy drop ratio, and non-dimensional speed, are used. This problem is demonstrated in Figure 25.

3.2.4.2 Characteristics for Determining Gas Properties

NUMBER OF MAPS REQUIRED

Without the use of a beta mapping variable, only two maps are required, neglecting second-order effects; with beta, three maps are needed. The use of a beta variable produces a cuboidal matrix, with consequent easier implementation in software.

REQUIREMENT FOR CONTINUITY

Unlike with a compressor, a discontinuity in the efficiency will not normally be observed. While it is possible for a turbine to behave as a stirrer or even as a compressor, this occurs at conditions which are unlikely to be achieved in practice. This can only occur at high rotational speeds and low flows. If the turbine is integrated into an engine, under such circumstances, the compressor will work well and thus absorb a considerable amount of energy. As this work is, under most conditions, supplied by the turbine, it follows that the turbine must be operating in turbine mode. The possible exception is when the engine shaft is being powered externally.

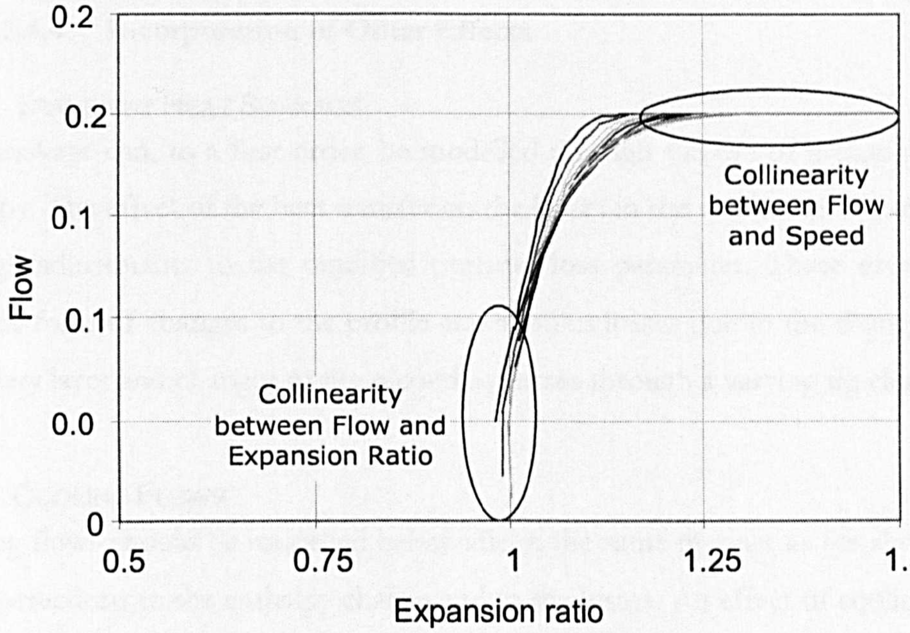

Figure 25 - Turbine Characteristic Plotted Using the Variables *Non-Dimensional Flow, Non-Dimensional Speed* and *Expansion Ratio* and Showing Collinearity Problems

DEFINITION FOR ZERO ROTATIONAL SPEED

The use of the variables specific torque and modified pressure loss permits the definition of zero speed behaviour.

3.2.4.3 General Requirements for Turbine Characteristics

LINEAR INDEPENDENCE

Without linear independence of all the mapped variables, the full set of flow conditions cannot be defined.

Characteristic Storage and Interpolation

The use of a beta mapping variable allows a cuboidal matrix to be formed for the maps. This is significantly easier to code than a matrix of non-uniform shape.

The variables must be smooth where possible, in order to allow reliable interpolation.

3.2.4.4 Incorporation of Other Effects

Transient Heat Soakage

Heat soakage can, to a first order, be modelled through the use of a change in the enthalpy. The effect of the heat transfer on the losses in the turbine can be modelled through adjustments to the modified pressure loss parameter. These extra losses take the form of changes to the profile and annulus losses due to the change in the boundary layer and changes to the secondary losses through a varying tip clearance.

Cooling Flows

Cooling flows should be modelled below-idle in the same manner as for above-idle, with corrections to the enthalpy change and to the losses. An effect of cooling flows is that the inlet and outlet mass flow rates are not normally equal. However, the flow rate in a turbine operating at low expansion ratios is determined by the flow area of the nozzle guide vanes (NGVs). Therefore, the value of flow rate used in the turbine characteristics should be that at the entry to the NGVs.

Reynolds Number Effects

During normal operation of a turbine, the temperature and pressure are high and so Reynolds number effects are negligible. However, during windmilling, this is no longer the case and so this should be considered. This is relatively easily incorporated into a characteristic using the variables of non-dimensional flow and modified pressure loss.

Inlet Temperature

The turbine experiences extreme temperature changes when going from idle running to windmilling. As the specific heat is a function of temperature, the values of c_p and gamma change. In some representation methods, this affects the performance of the turbine. However, for the method using non-dimensional flow, specific torque, modified pressure loss, non-dimensional speed and beta, the representation is believed to hold.

Working Fluid

During normal operation, the working fluid in the turbine is the products of combustion. However, when windmilling, the working fluid changes to air. These two fluids have different gas constants, although the change is relatively small. Most representation methods do not allow for this change. However, the method being proposed is believed to produce valid results using the alternative working fluid.

Volume Packing

During a rapid acceleration or deceleration, the working fluid changes its density. Therefore, it is possible to have a situation whereby the flow into a component is not equal to the flow out of it, due to the density change or volume packing effect. This is easily corrected for using the variable of non-dimensional mass flow rate.

3.2.4.5 Conclusions

Matching Parameters

Corrected speed is invariably used as one of the two parameters necessary to define the operating point of the turbine. A further variable is then required. The main options for this variable are described in Table 4.

Table 4 - Matching Parameters for Turbine Characteristics

Matching parameter	Comment
Expansion ratio	Multiple solutions at low speeds and flows. Inlet temperature dependent.
Q_{in}	Multiple solutions at high flows.
β	Uniquely defines operating point. Independent of inlet temperature. Requires three look-up tables rather than two.

INDEPENDENT VARIABLES

When β is used as the matching parameter, three independent variables are required to determine the flow conditions. One of these is invariably the inlet flow function, Q_{in}. Table 5 looks at the remaining possible variables.

Table 5 - Independent Variables for Turbine Characteristic Representation

Independent variables	Comments
Π, η	Pressure ratio dependent on inlet temperature. Efficiency does not remain finite and is not independent of inlet temperature.
$\frac{\Delta H}{T_{in}}, \frac{\Delta H'}{T_{in}}$	Zero-speed torque not defined. Ideal enthalpy rise not independent of inlet temperature.
τ_{spec}, P_{loss}	Zero-speed torque defined. Pressure loss not independent of inlet temperature.
τ_{spec}, P_{loss}^*	Zero-speed torque defined. Modified pressure loss largely independent of inlet temperature.

RECOMMENDATIONS

The variables of beta and non-dimensional speed are suitable for defining the operating point of the turbine. These create a uniform look-up table, resulting in a simpler style of programming and more reliable values of iterative guesses. The independent variables to be recommended are the non-dimensional mass flow rate, the specific torque and the modified pressure loss. These variables are believed to be independent of changes in working fluid, a phenomenon which the turbine experiences in going from idle running to windmilling.

Characteristics resultant from this choice of variables are shown in Figure 26, Figure 27 and Figure 28.

3.2.5 Effect of Representations on Predicted Engine Performance

3.2.5.1 Introduction

In order to quantify the effect of a change of representation method, an overall gas turbine performance model was assembled, as described in Chapter 4. This was then run using each of the characteristic representation methods, as shown in Table 6. The results of this analysis are presented in this section.

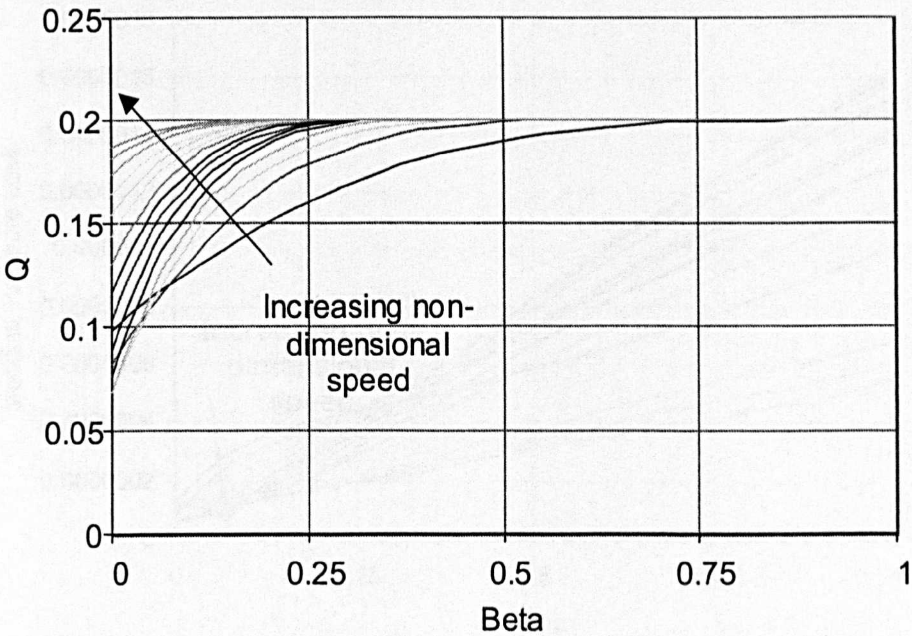

Figure 26 - Turbine Characteristic Showing Variation of *Flow Function* with *Beta* for a Range of *Non-Dimensional Speeds*

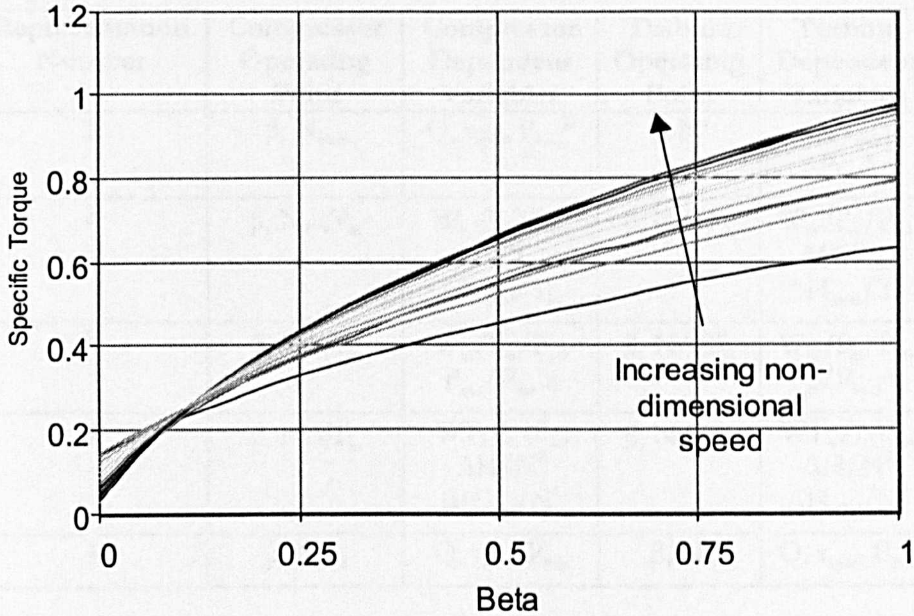

Figure 27 - Turbine Characteristic Showing Variation of *Specific Torque* with *Beta* for a Range of *Non-Dimensional Speeds*

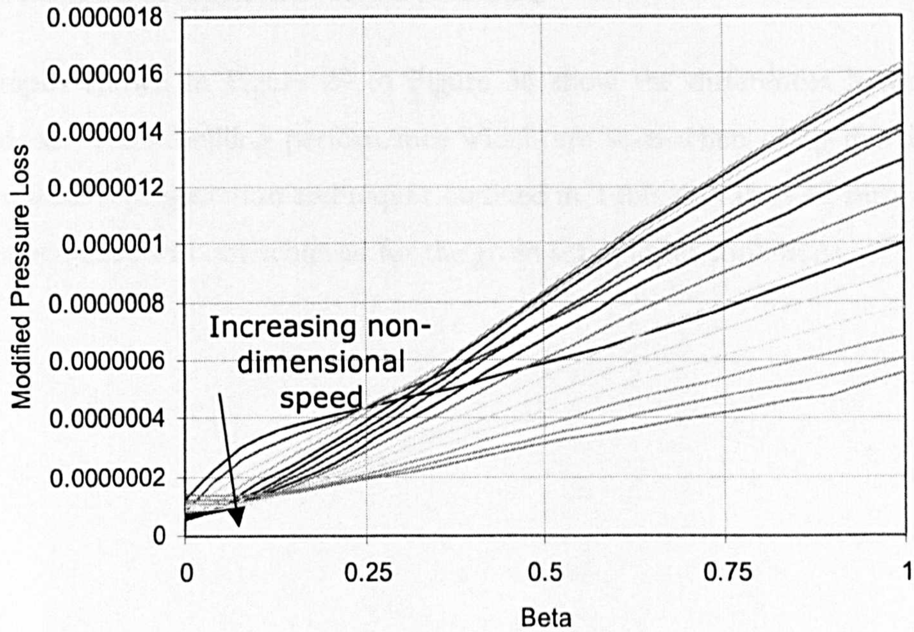

Figure 28 - Turbine Characteristic Showing Variation of *Modified Pressure Loss* with *Beta* for a Range of *Non-Dimensional Speeds*

Table 6 - Turbomachinery Characteristic Representation Methods Tested

Representation Number	Compressor Operating Point	Compressor Dependent Variables	Turbine Operating Point	Turbine Dependent Variables
1	β, N_{corr}	Q, τ_{spec}, P_{loss}*	β, N'	Q, τ_{spec}, P_{loss}*
2	β, $N/\sqrt{T_{in}}$	$W_{in}\sqrt{T_{in}}/P_{in}$, $\Delta H/T_{in}$, $\Delta H_{ideal}/T_{in}$	β, $N/\sqrt{T_{in}}$	$W_{in}\sqrt{T_{in}}/P_{in}$, $\Delta H/T_{in}$, $\Delta H_{ideal}/T_{in}$
3	β, $N/\sqrt{T_{in}}$	$W_{in}\sqrt{T_{in}}/P_{in}$, P_{out}/P_{in}, η	β, $N/\sqrt{T_{in}}$	$W_{in}\sqrt{T_{in}}/P_{in}$, P_{in}/P_{out}, η
4	β, $N/\sqrt{T_{in}}$	WT_{in}/NP_{in}, $\Delta H/N^2$, $\Delta H_{ideal}/N^2$	β, $N/\sqrt{T_{in}}$	WT_{in}/NP_{in}, $\Delta H/N^2$, $\Delta H_{ideal}/N^2$
5	β, N_{corr}	Q, τ_{spec}, P_{loss}	β, N'	Q, τ_{spec}, P_{loss}

3.2.5.2 Results

In this section, the differences in the performance synthesis results between the various methods of representation are presented.

The graphs shown in Figure 29 to Figure 36 show the differences between the predictions of windmilling performance which are seen when using the different characteristic representation techniques outlined in Table 6. Values of zero indicate that convergence was not achieved for the given set of input conditions.

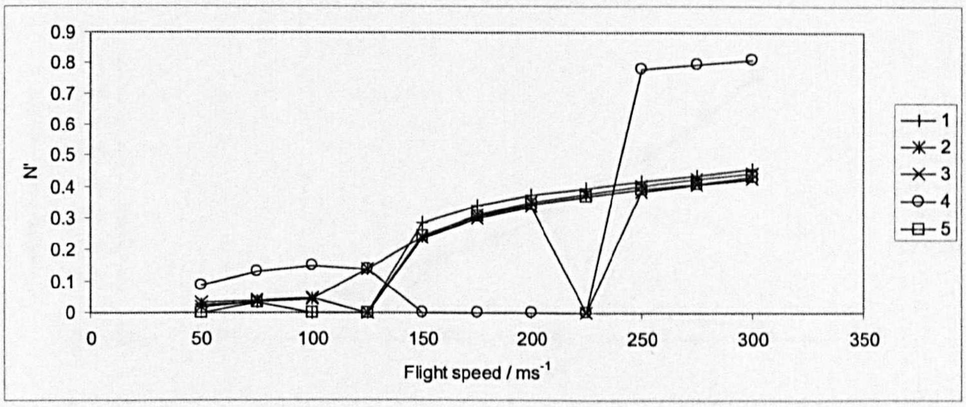

Figure 29 - Effect of Choice of Characteristic Representation Method on Windmilling Performance Prediction of *Non-Dimensional Spool Speed* for a Simulated Altitude of 5000 m (16500 ft) at Varying Flight Speeds

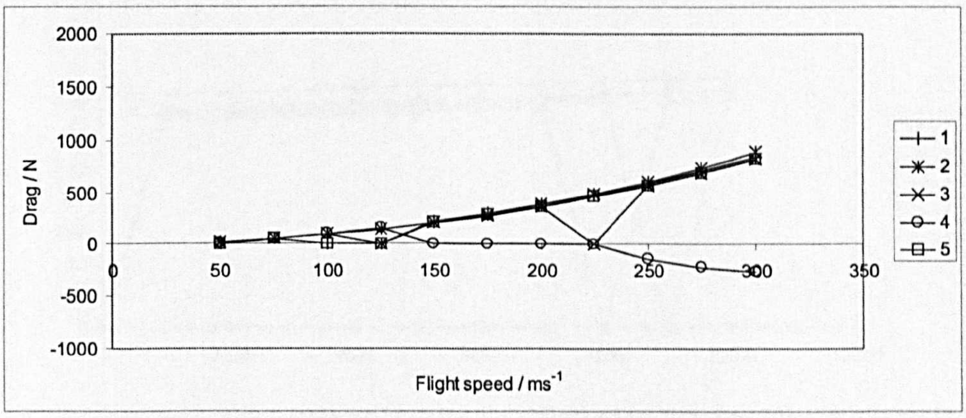

Figure 30 - Effect of Choice of Characteristic Representation Method on Windmilling Performance Prediction of *Internal Drag* for a Simulated Altitude of 5000 m (16500 ft) at Varying Flight Speeds

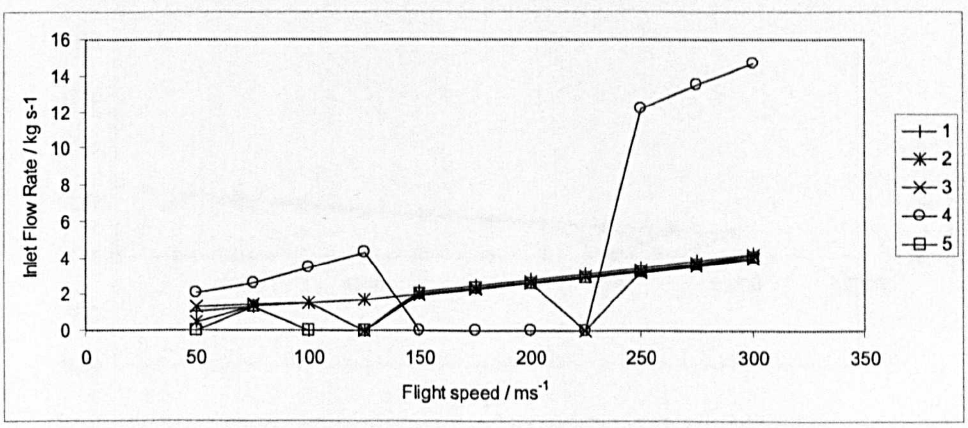

Figure 31 - Effect of Choice of Characteristic Representation Method on Windmilling Performance Prediction of *Engine Mass Flow Rate* for a Simulated Altitude of 5000 m (16500 ft) at Varying Flight Speeds

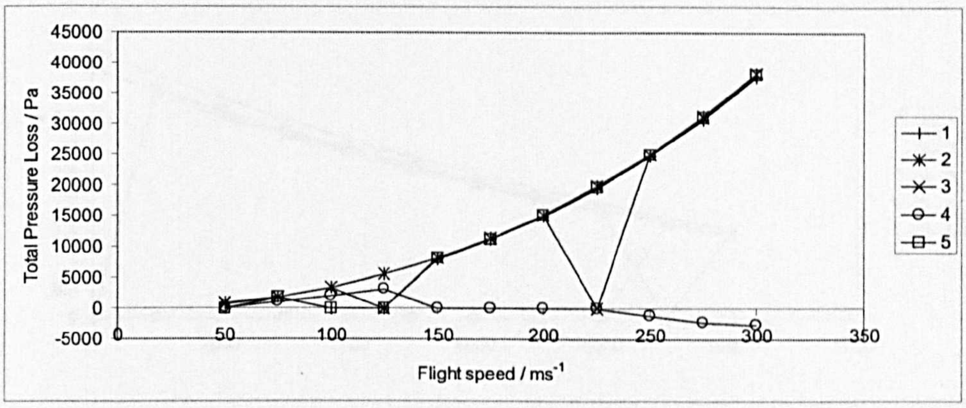

Figure 32 - Effect of Choice of Characteristic Representation Method on Windmilling Performance Prediction of *Loss in Total Pressure* from Intake to Nozzle Exit for a Simulated Altitude of 5000 m (16500 ft) at Varying Flight Speeds

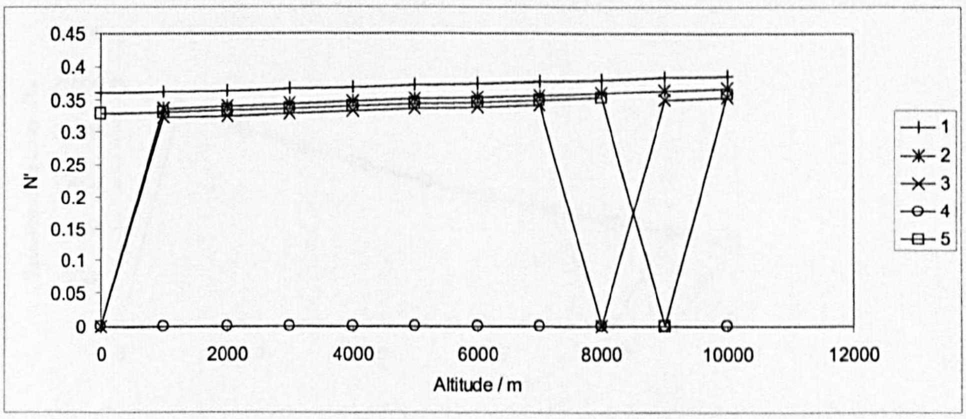

Figure 33 - Effect of Choice of Characteristic Representation Method on Windmilling Performance Prediction of *Non-Dimensional Spool Speed* for a Simulated Flight Speed of 200 ms^{-1} at Varying Altitudes

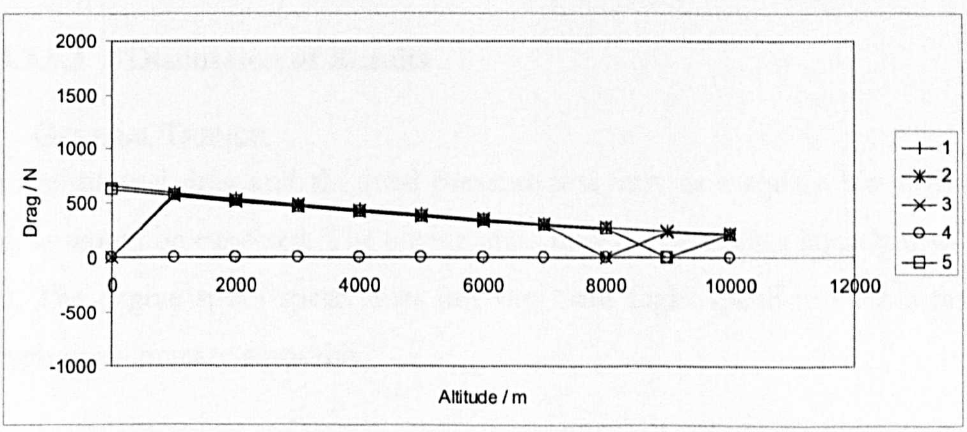

Figure 34 - Effect of Choice of Characteristic Representation Method on Windmilling Performance Prediction of *Internal Drag* for a Simulated Flight Speed of 200 ms^{-1} at Varying Altitudes

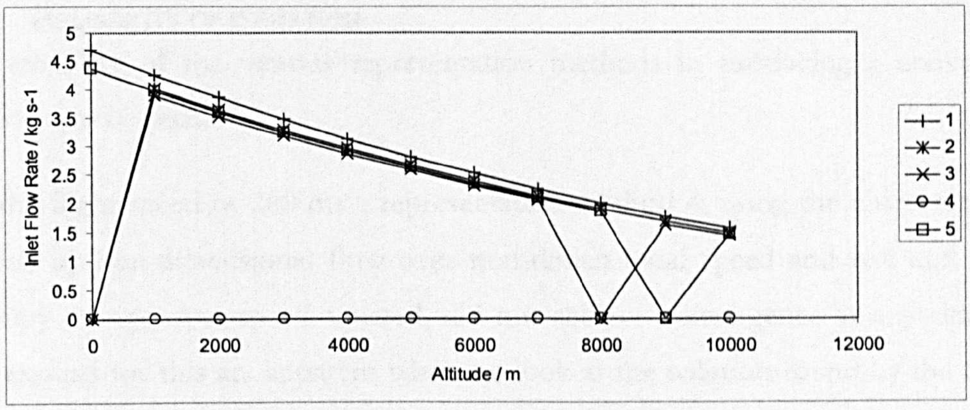

Figure 35 - Effect of Choice of Characteristic Representation Method on Windmilling Performance Prediction of *Engine Mass Flow Rate* for a Simulated Flight Speed of 200 ms^{-1} at Varying Altitudes

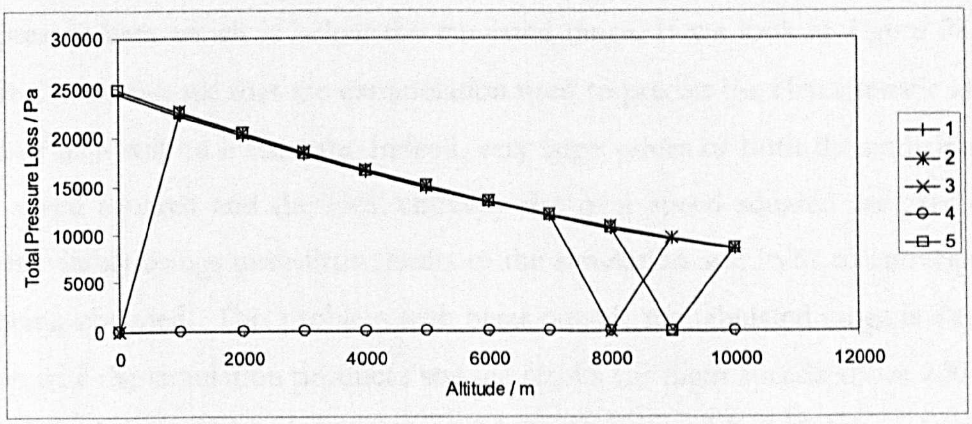

Figure 36 - Effect of Choice of Characteristic Representation Method on Windmilling Performance Prediction of *Loss in Total Pressure* from Intake to Nozzle Exit for a Simulated Flight Speed of 200 ms^{-1} at Varying Altitudes

3.2.5.3 Discussion of Results

General Trends

Both the internal drag and the total pressure loss vary as a square law with flight speed, as would be expected. The engine mass flow rate increases linearly with flight speed. The engine spool speed does not vary with flight speed in such a fashion, although does increase smoothly.

With increasing altitude, the engine rotational speed increases slightly at constant flight speed. The internal drag, air flow rate and pressure loss decrease with increasing altitude.

RELIABILITY OF ITERATION

The reliability of the various representation methods in producing a converged solution can be seen.

For the flight speed of 200 ms^{-1}, representation method 4, using the characteristics defined by non-dimensional flow over non-dimensional speed and real and ideal enthalpy change over speed squared, did not achieve convergence at any altitude. The reasons for this are apparent when we look at the solution found by the other representation methods and the compressor characteristic as defined by the variables enthalpy rise over speed squared and ideal enthalpy rise over speed squared. As can be seen in Figure 37, the solution for this point falls at a compressor beta which is below the tabulated range. If we look at Figure 38 and Figure 39, we can see that the extrapolation used to predict the characteristic at this value of beta will be inaccurate. Indeed, very large values of both the enthalpy rise over speed squared and the ideal enthalpy rise over speed squared are predicted. This inevitably brings unrealistic results to the simulation and leads to convergence not being obtained. This problem with betas outside the tabulated range is also the reason why the simulation produces strange results for flight speeds above 200 ms^{-1} at 5000 m altitude. Here, the compressor defies the second law of thermodynamics due to a poor extrapolation. Clearly, this method of representation is not to be recommended.

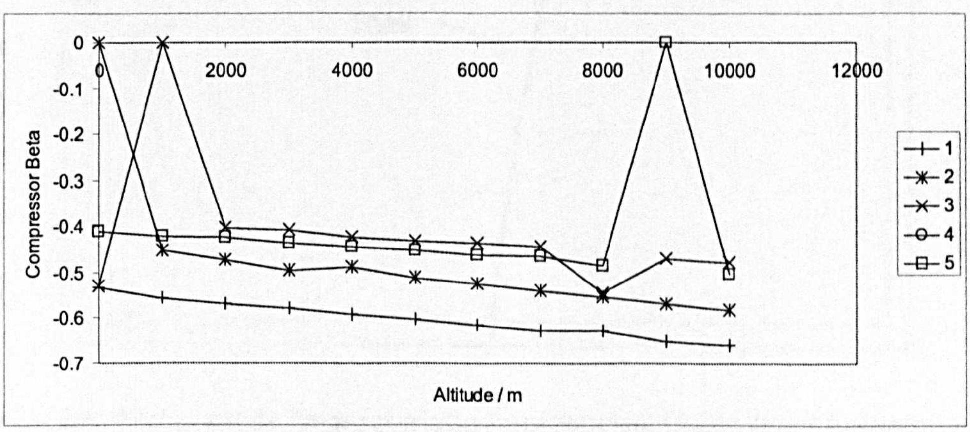

Figure 37 - Variation of Compressor Beta with Altitude for a Flight Speed of 200 ms^{-1} for the different representation Techniques

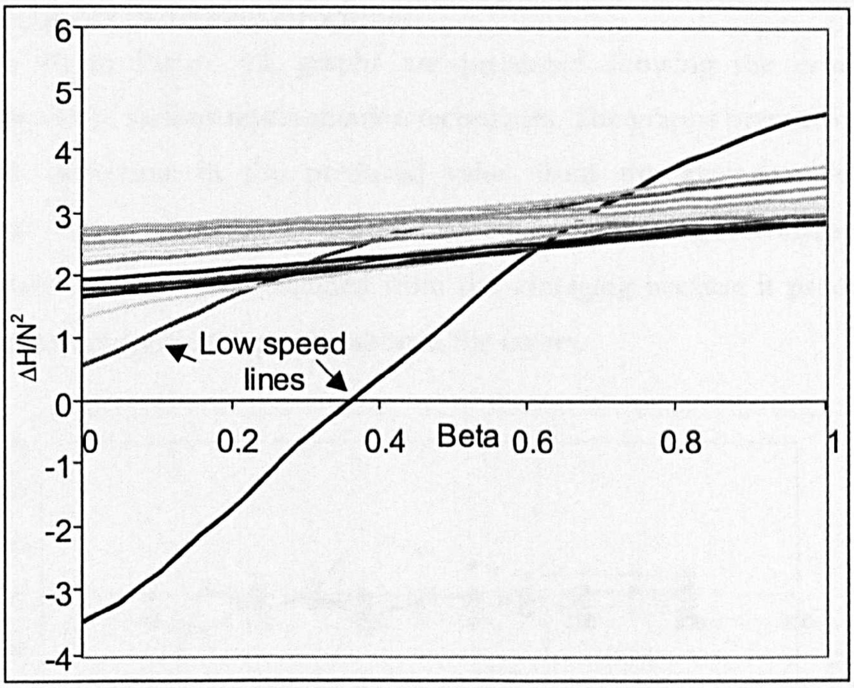

Figure 38 - Compressor Characteristic Represented Using *Enthalpy Rise over Speed Squared*

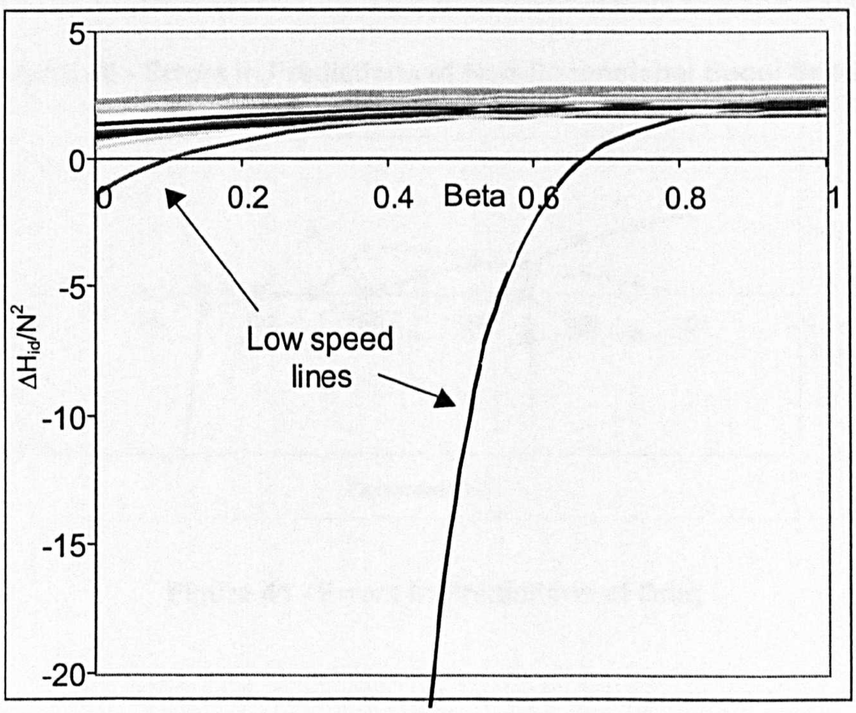

Figure 39 - Compressor Characteristic Represented Using *Ideal Enthalpy Rise over Speed Squared*

ACCURACY OF REPRESENTATIONS

In Figure 40 to Figure 43, graphs are presented showing the errors in the predictions of the various representation techniques. The graphs presented show the percentage difference in the predicted value from the average value of the predictions across the representation techniques, excepting technique 4. This representation method was excluded from the averaging because it produced very different answers and consequently skewed the errors.

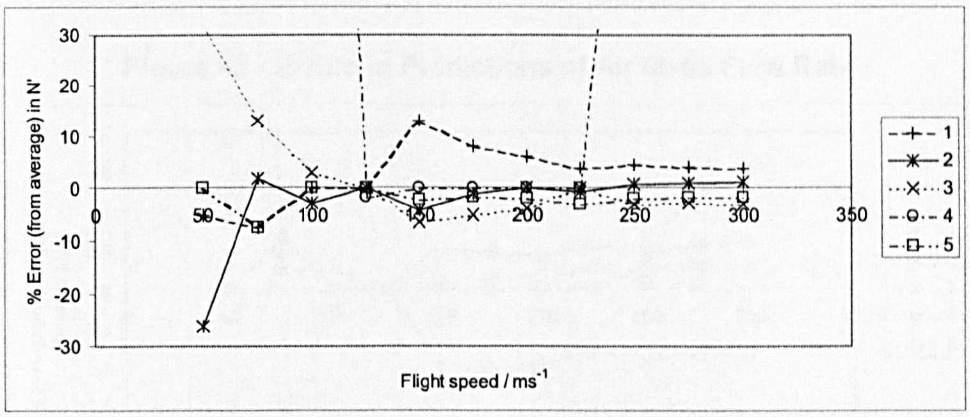

Figure 40 - Errors in Predictions of Non-Dimensional Spool Speed

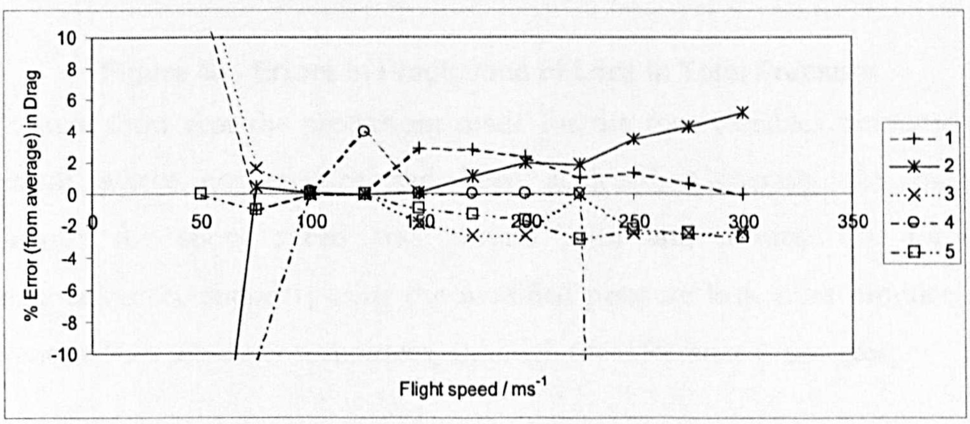

Figure 41 - Errors in Predictions of Drag

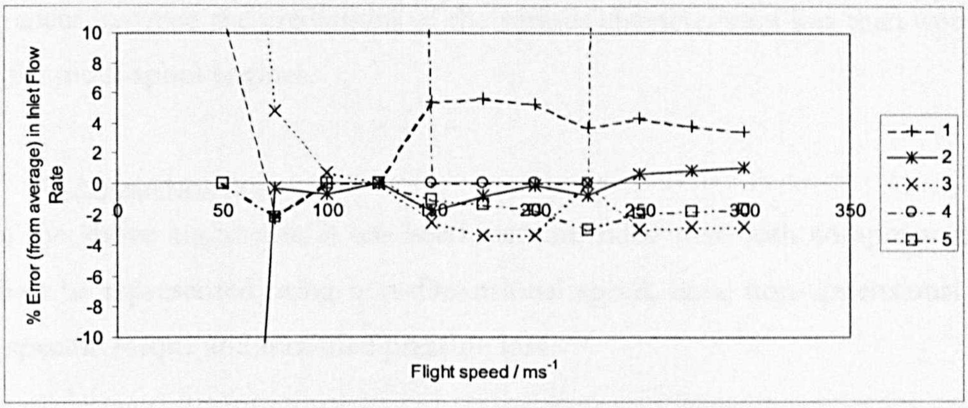

Figure 42 - Errors in Predictions of Air Mass Flow Rate

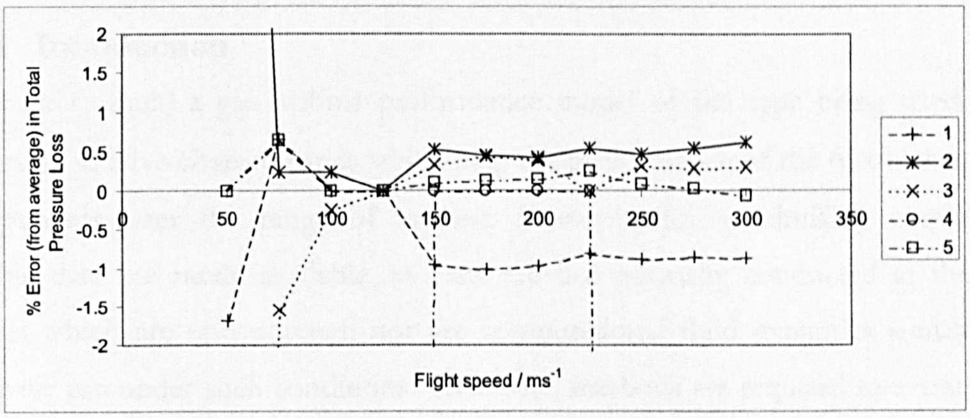

Figure 43 - Errors in Predictions of Loss in Total Pressure

The results show that the predictions made for the four variables examined were consistent where convergence had been achieved. Generally, the range of predictions for spool speed was around 10% and around 6% for drag. Representation technique 1, using the modified pressure loss, does produce some differences from the other techniques, although the difference is not great.

MULTI-SPOOL ENGINES

Simulations have not been conducted on multi-spool engines. However, as the variation in temperature affects a larger number of components, greater differences between the representations would be expected. The use of a single-spool engine in this simulation means that the compressor system does not experience a temperature change between normal flight operation and windmilling, resulting in differences between the predictions of the various characteristics less than would be seen for multi-spool engines.

RECOMMENDATION

From the above arguments, it has been recommended that both compressors and turbines be represented using non-dimensional speed, beta, non-dimensional flow rate, specific torque and modified pressure loss.

3.3 TURBOMACHINERY CHARACTERISTIC EXTRAPOLATION

3.3.1 Introduction

In order to build a gas turbine performance model of the type being used, it is necessary to have characteristics which map the performance of the turbomachinery components over the range of interest. However, for windmilling conditions, reliable data are rarely available, as tests are not normally conducted at the low speeds which are encountered, nor are computational fluid dynamics simulations normally run under such conditions. Therefore, methods are required to extrapolate the above-idle characteristics to the speeds found during windmilling. This section describes, first for compressors and then for turbines, methods of performing such an extrapolation.

3.3.2 Compressor Characteristic Extrapolation

3.3.2.1 Introduction

Typically, compressor performance data are only available for speeds above around 50% of design point. As the windmilling regime frequently sees engine speeds substantially lower than this, it is necessary to determine the performance of the compressor outside the range for which information is typically available.

A paper was presented on this topic by Jones et al. [34] at Turbo Expo 2002 as part of this doctorate. The contents of this paper are discussed in more detail here.

3.3.2.2 Issues Encountered In Extrapolating Compressor Characteristics

Low Speed Compressor Operation

When operating at low speeds, a compressor may operate in any of three modes. At low flow speeds, the compressor will behave normally, giving energy to the airstream and producing a pressure rise. At higher flow speeds, the losses in the compressor become more dominant. This results in a stirrer mode of operation, where energy is still being given to the airstream, but a pressure loss is seen due to the losses. The final mode of operation is where the airflow is high and the compressor now acts in a turbine mode, taking energy from the airstream.

Discontinuities in Variables

In Section 3.3.3.2, the problems associated with variables which become discontinuous during certain operating conditions are described. Particularly, it is noted that isentropic efficiency experiences a discontinuity as the compressor moves between the stirrer and turbine modes of operation.

Enthalpy Changes

The change in enthalpy across a stage of a compressor can be determined analytically by Euler's equation (Eq. 6).

Eq. 6 $$\frac{\Delta H}{U^2}=\frac{\Delta Vw}{U}=\frac{Va}{U}\left(\tan\alpha_3-\tan\alpha_0\right)\Rightarrow\frac{\Delta H}{N^2}\propto\frac{WT}{NP}\left(\tan\alpha_3-\tan\alpha_0\right)$$

However, this relationship is based around air angles. Particularly at incidences far from the design incidence, these air angles may not be similar to the blade angles, and therefore the enthalpy rise across the stage may not be determined using a purely analytical technique.

Losses

The losses in a compressor come from three main sources. These are often described as the annulus loss, profile losses and secondary losses. The annulus loss can be predicted to some extent by the use of a skin friction relationship. The secondary losses are generally going to be small at low pressure rises for high Va/U. The profile losses, however, are not easily predicted due to the changes in incidence angles which may be experienced in the sub-idle region.

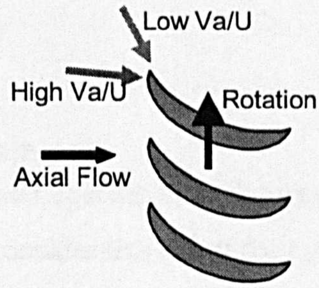

Figure 44 - Large angles of incidence at low speed operation of a compressor rotor

During above-idle operation of a compressor, high values of Va/U are not encountered, due to the choking of the compressor. However, they may be seen at low speed, as the choking will then not occur. As the profile drag will be heavily dependent upon the angle of incidence, this will then be difficult to predict when we have no experimental data from this regime. The region in which an extrapolation is thus uncertain is shown in Figure 45.

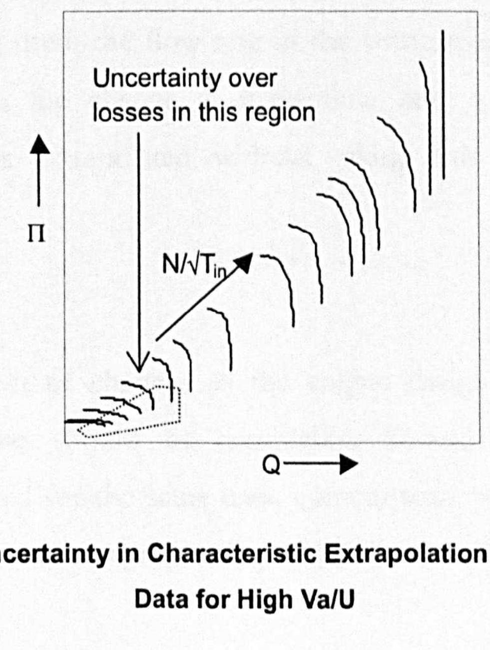

Figure 45 - Region of Uncertainty in Characteristic Extrapolation due to Insufficient Loss Data for High Va/U

One method around this is however presented by Choi et al. [13]. This is to use the aerodynamic relationship that the drag coefficient is proportional to the square of the lift coefficient for an aerofoil. As the lift coefficient can approximately be determined by the enthalpy rise across the stage if the compressor geometry is known, it should therefore be possible to calculate an approximate value for the pressure loss.

Specific Heat Changes

While it is important to consider specific heat changes in a gas turbine performance model, it is not necessary to consider this effect for extrapolation of characteristics.

Reynolds Number Effects

Reynolds number effects influence the behaviour of a compressor at low speeds of operation.

Bleeds

As bleeds open and close, the internal matching of the compressor will alter. Should any of these bleeds therefore operate at speeds below those for which data are available, this may present problems for the extrapolation.

When a bleed is being used, the flow rate in the compressor will not be constant along its length. With the change in mass flow rate, a complete compressor characteristic which is extrapolated without taking this into account will be incorrect.

Repeatability

In order that the effect of changes to the engine design can be modelled, the extrapolation technique should be repeatable. Should different extrapolated characteristics be created for the same data, comparisons between different engine designs and layouts will not be completely possible.

TEST TO SHOW WHETHER NEGATIVE INCIDENCE DATA ARE AVAILABLE

As extrapolation of a compressor characteristic to below-idle conditions is dependent on the availability of unchoked negative incidence data, one must first check whether such data are available. In order to do this, we use the backbone method, described below. By overlaying the minimum loss backbone on our known characteristic, one can see whether unchoked negative incidence data are available.

3.3.2.3 Analysis of Various Compressor Characteristic Extrapolation Techniques

YAN & MAI [85]

A technique for extrapolating compressor characteristics through a stage-stacking approach is presented by Yan and Zhong Fan [85].

The method approximates the stage characteristics using polynomials. These polynomials are then extrapolated.

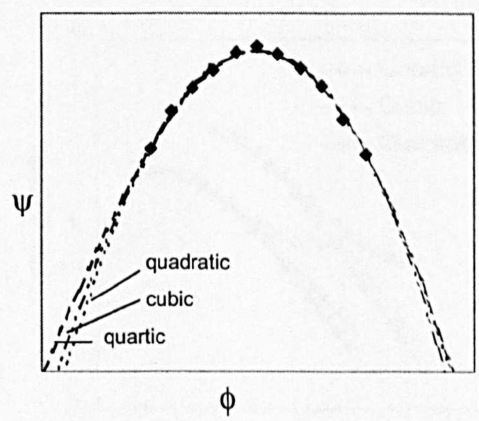

Figure 46 - Fitting Polynomials of Different Degrees to Stage Data

As with any polynomial fitting method, the function may be expected to give results reasonably close to the actual results within the range which has been studied, but may not give reliable results outside that range. This is shown in Figure 46, where polynomials of the second, third and fourth orders are fitted to a set of data points. Within the range for which we have data, the results found using the polynomials are very close to those we would expect. However, outside the range of known data, the polynomials produce different results.

Furthermore, approximation of stage characteristics through the use of extrapolated polynomials will not model the effect of stall. Stall can be encountered at both high and low values of flow coefficient and greatly influences the overall compressor characteristic. Therefore, this technique will produce inaccurate characteristics when any stage approaches a stall condition.

When stage characteristics described using polynomials of different orders are stacked to produce a characteristic for the overall compressor, the different results are very apparent. Figure 47 shows two low speed lines on a compressor characteristic which have been created through stacking together three stages whose performances have been described by polynomials of second, third and fourth order as shown in Figure 46.

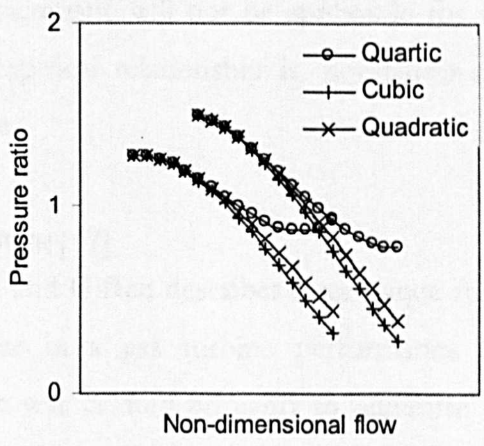

Figure 47 - Two Low Speed Lines of a Compressor Characteristic, Determined by Stacking Three Compressor Stages with Performance Described by Polynomials of 2nd, 3rd and 4th Order

Data on a stage characteristic will normally only be known for the medium to low flow coefficient region. However, characteristics for windmilling and starting are required to be reliable for high and very low flow coefficients. In these regimes, the flow will tend to separate, leading to stage characteristics with large changes in gradient outside the region for which data are available.

These two factors, of the inherent inaccuracy of extrapolating polynomials, and the effect of separation on stage characteristics, lead to a problem with the method. Nevertheless, it can provide a good basis for building a model.

AGRAWAL & YUNIS [4]

Agrawal & Yunis [4] describe the performance of the compressor as a generalised running line for start-up. Based on data from a number of engines, empirical relationships are found to determine the flow, enthalpy rise and efficiency from the speed.

This approach is sufficient to describe much of the start-up operation of the gas turbine engine, with the shaft torque and the compressor exit temperature and pressure available as output. However, since only the start-up running line is considered, such a characteristic will not be applicable for windmilling modelling. The use of such an empirical relationship is, nevertheless, a reasonably reliable approach in this instance.

CONVERSE & GIFFEN [17]

The report of Converse and Giffen describes a technique for the representation of compressor performance in a gas turbine performance simulation deck. This representation technique was created primarily to minimise the computer memory required for holding the compressor map while maintaining its precision. However, it may also form the basis of a technique to extrapolate compressor characteristics.

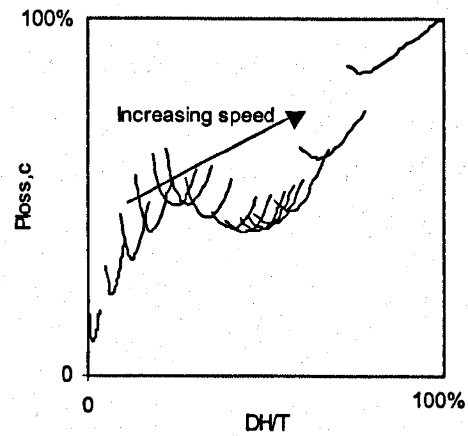

Figure 48 - Map Showing Variation of Pressure Loss with Work Coefficient

First, the compressor characteristic is plotted showing the pressure loss against the work coefficient for lines of constant non-dimensional speed. Such a map is shown in Figure 48.

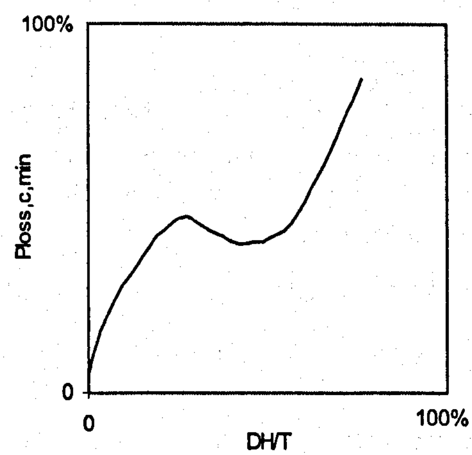

Figure 49 - Minimum Loss Backbone of Minimum Pressure Losses against Work Coefficient

From this chart, the minimum loss points are found and plotted to form a backbone. Such a map is shown in Figure 49. It should then be possible to extrapolate the curve to give the minimum pressure loss for any value of enthalpy rise and hence speed.

The difference of pressure loss from the minimum value is then plotted for each speed as in Figure 50. In the report, the authors then linearise Figure 50 in order to minimise computer memory, although this computing requirement is not as much of an issue with the more powerful computers of today. The pressure losses away from the minimum loss may then be predicted for low speeds by studying the variation of pressure losses at higher speeds.

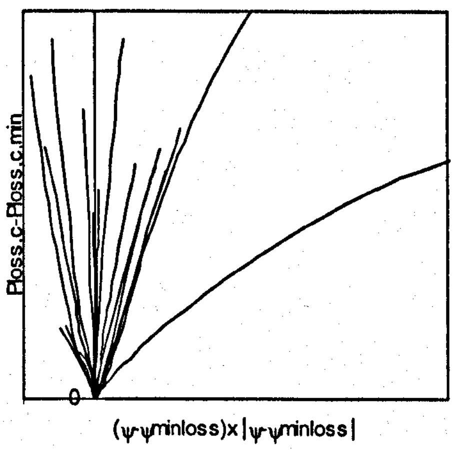

Figure 50 - Map Showing the Pressure Loss Variation from Minimum Loss for Lines of Constant Non-Dimensional Speed against the Square of Work Coefficient

The maps of Figure 49 and Figure 50 thus describe the pressure losses in the compressor, a measure of its efficiency, in terms of the work coefficient and the non-dimensional speed.

One problem with this technique when applied to the problem of extrapolation is that, with data generally only available from around 60% of design speed upwards, the kink seen on Figure 49 will not be predicted. Such kinks are due to the operation of variable geometry and bleed valves, and thus could be predicted by other means.

RIEGLER ET AL. [66]

In their paper of 2000 [66], Riegler et al. use the software SmoothC to perform the extrapolation. The software is described in the paper of Kurzke [40]. The extrapolation uses the known operating point of zero-speed and zero-flow to reduce the uncertainty. At these conditions, the enthalpy rise is zero, the torque is zero, the pressure ratio is unity and the gradient of the zero-speed pressure ratio curve is zero.

Riegler et al. then use the software to extrapolate graphically the above-idle data on the bases that speed lines on a plot of ψ_s versus ϕ come close together at low speeds, that low speed lines on a specific torque versus non-dimensional flow plot are nearly parallel and that the low-speed peak efficiency line plotted versus non-dimensional flow or speed becomes horizontal. The authors note that, while this technique is reasonably reliable for angles of incidence near design point, the use of engineering judgement and a largely graphical technique produce inconsistent results as Va/U moves to high values.

3.3.2.4 Alternative Potential Techniques

ZERO SPEED LOSS AND TORQUE CURVES

Extrapolation is inevitably less reliable than interpolation. Therefore, if the zero speed loss and torque curves could be defined, these would, through providing a lower bound to the extrapolation, probably provide better results than a technique purely using extrapolation.

The zero speed loss and torque curves could be provided by three possible techniques:

- Calculation
- Measurement
- Estimation

Calculation

Through the use of Computational Fluid Dynamics, it should be possible to provide a zero speed loss curve for the compressor system. However, a number of problems are encountered in trying to perform such an analysis.

As with any CFD loss model, some calibration and validation of the results is required. As data are not normally available against which the computed results may be compared, this will not generally be possible.

The detail of geometry required in the analysis is quite high in order to predict the separation which will occur at high angles of incidence. Furthermore, a single stage model of the compressor will not be sufficient, as there will be radial non-uniformity of flow at entry to stages after the first, and the reducing annulus area will have a large effect on the characteristic, as the rear stages become compressible. This then tends to create a geometry which is very demanding in terms of the computer memory and processing power required to model it.

The time required to set up such a computer simulation would be large. It would also require the use of data on compressor geometry which the performance analysts would not normally be required to use.

This technique has been investigated further, as part of this work, by an MSc student, Romain Chambard [11] and is described in Section 3.3.2.8.

Measurement

The mapping of the zero speed loss line of a compressor map would require considerably less testing time than the mapping of the entire compressor characteristic in the sub-idle regime. However, it would still require an alternative design of compressor test rig in order to create a pressure drop across the compressor and to lock the rotor, rather than providing power to it. Furthermore, instrumentation designed for above-idle testing will be inappropriate for sub-idle tests, where the temperatures and pressures encountered are substantially different from those at design point and the measurements must be made to a high accuracy. The test would, nevertheless, be simpler than a full sub-idle test, as the use of a braking system which can act while the compressor shaft is rotating is not required.

This technique is also suggested by Kurzke [40].

Estimation
If drag coefficients for the blades could be estimated, the pressure losses could then be estimated by totalling the losses across each stage using a simple drag coefficient relationship. This technique would also require the use of data which performance analysts do not normally need, but would be a relatively simple calculation.

AN ALTERNATIVE STAGE-BY-STAGE EXTRAPOLATION TECHNIQUE
Yan and Mai [85] present a technique in which the stage characteristics of a compressor are extrapolated and then stacked to form an overall characteristic. However, this technique requires a knowledge of the stage characteristics, information which will not normally be readily available.

A number of empirical relationships exist for predicting the performance of a compressor stage. These are presented in a variety of texts, including Horlock [27]. As with most empirical relationships, these generally include a number of constants which are not known. However, through guessing these constants and stacking the compressor, the resultant map may be compared with the original. Using an error minimisation technique, the constants may then be varied until the stacked map is very close to the original. Once this has been achieved, the stage characteristics may be extrapolated using the empirical relationships. The restacked overall compressor map should then be close to the true low-speed characteristic.

This method is explored further in Section 3.3.4.

GENERALISED CHARACTERISTICS
In the conceptional design of engines, generalised characteristics are often employed. These characteristics are then scaled to produce the correct design point parameters.

One approach to the problem of below-idle modelling is to use experimental data for the above-idle regime, and to use a scaled generalised characteristic for below-idle conditions.

A number of techniques exist for scaling compressor characteristics. Some of these are described in the paper of Kurzke [39].

In scaling the sub-idle characteristic, a point must be chosen at which to perform the scaling. This introduces an element of engineering judgement, thus limiting the repeatability.

The use of generalised characteristics, based on experimental data or computed results from a number of engines, should produce reliable results, though not as good as those which are extrapolated well.

3.3.2.5 Variable Geometry and Bleeds

The extrapolation of the characteristic of a compressor with variable geometry or bleed flows presents some problems for those techniques which do not use a stage-by-stage analysis. These problems can be overcome through the extrapolation of characteristics for various settings of the variable geometry devices and a bleed schedule. However, particularly in projects between a variety of companies, where such information may be protected, or for modelling by engineers outside an engine manufacturing company, this information may not be available. However, if significant quantities of data are available from experiment for compressor behaviour below the operating points of the bleeds and variable geometry, the above techniques should all be reasonably good in this respect.

3.3.2.6 Comparison Between the Techniques

The above techniques will all produce slightly different results.

The technique of Agrawal and Yunis [4] uses the compressor efficiency in creating the characteristic. As described in Jones et al. [33], efficiency experiences a discontinuity as the compressor transitions from stirrer to turbine modes of operation.

All the techniques described use an enthalpy rise parameter in the characteristic definition. As this is a result of Euler's equation, this is recommended. The use of specific torque is preferable to enthalpy rise, as it describes the torque at zero rotational speed.

With regard to pressure losses, only the zero speed loss curve and alternative stage-by-stage technique take into account the effects of high pressure losses at large angles of incidence.

Due to the effects of bleeds, variable geometry and other factors, most of the techniques require some engineering judgement in the extrapolation. This inevitably leads to some variability in the results produced. Those techniques which investigate stage performances are more reliable in this respect. Also, the use of a zero-speed pressure loss and torque characteristic will result in far more repeatable extrapolation.

The generating of a zero-speed characteristic is generally difficult. However, the extra reliability gained by this technique will generally benefit the resultant characteristics enormously.

In order to generate a zero-speed map through computational methods, detailed knowledge of the compressor geometry is required. This is also true of the stage-by-stage approaches. However, the geometry may be estimated through the use of simple compressor design techniques applied to the design point of the compressor. Thus, simple blading will be generated, from which the calculations may be performed.

For most starting calculations, the method of Agrawal and Yunis [4] is sufficient. However, for windmilling calculations, where the running line may change for different conditions of flight speed and altitude, full characteristics are required.

3.3.2.7 Conclusion

A number of techniques for extrapolating compressor characteristics to the sub-idle region have been described.

For starting studies, the technique of Agrawal and Yunis [4] will normally be sufficient.

The method of Converse and Giffen [17] provides a good method of describing the compressor behaviour throughout the entire operating range of the component. However, the kink in the minimum loss backbone is difficult to predict using this method, and thus the potential as a technique for extrapolation is slightly limited.

The use of the software SmoothC [66],[40] produces good characteristics, based upon physical phenomena. However, it's reliance on engineering judgement means that results are not entirely reproducible.

The use of a zero-speed loss curve vastly increases the reliability of most extrapolation methods. However, its generation is difficult, either through experiment or computational methods.

The alternative stage-by-stage extrapolation method requires more development. However, the technique shows great promise for producing characteristics which are based on physical phenomena and are reproducible.

The use of scaled generalised characteristics will produce results which are realistic, but requires some engineering judgement in deciding at which conditions to perform the scaling and thus the results will not be completely reproducible.

3.3.2.8 Generation of a Zero-Speed Compressor Characteristic [11]

INTRODUCTION

The following work was performed by Romain Chambard, an M.Sc. student at Cranfield University, supervised by Dr. Pilidis and managed by the author of this thesis.

Following the recommendations of the previous section, it was decided that an investigation would be made into the feasibility of generating zero-speed loss and torque curves from Computational Fluid Dynamics (CFD).

METHOD

Chambard found that the CFD analysis should be performed using only a two-dimensional analysis, as this is far less computationally demanding than a full three-dimensional analysis. However, as one of the important phenomena in the flow through a compressor is the fact that the annulus area reduces along the length of the component, a simple two-dimensional representation would not suffice. Furthermore, the fact that the number of blades in a row was not constant along the compressor length meant that modelling the flow through more than one blade passage would be necessary in a true two-dimensional model. Finally, when studying a locked rotor condition, different results would be obtained with the rotor at a different position relative to the stator.

In order to address the first of the above issues, Chambard decided to thicken the blades of the rear stages of the compressor. Thus, while the profile was still the same basic shape as the actual compressor being studied, the thickness was not. The blade thickness was set to such a value as to create the same size blade passage as in the real compressor.

Addressing the second issue, he used a mixing plane model. In this, the flow through only a single blade passage per blade row was studied. The entry to the next stage carried the same average velocity of flow, temperature and pressure as the exit of the prior stage. However, mass flow continuity was not conserved as the flow areas were different.

The final issue was addressed by allowing the rotor to rotate, albeit at a very low speed, such that it could still be regarded as being at locked rotor conditions.

RESULTS

Surprisingly good results were obtained. These appeared to fit well with the data measured from a locked rotor test performed on the seven-stage compressor of an actual engine by Rolls-Royce. These results are presented in Figure 51.

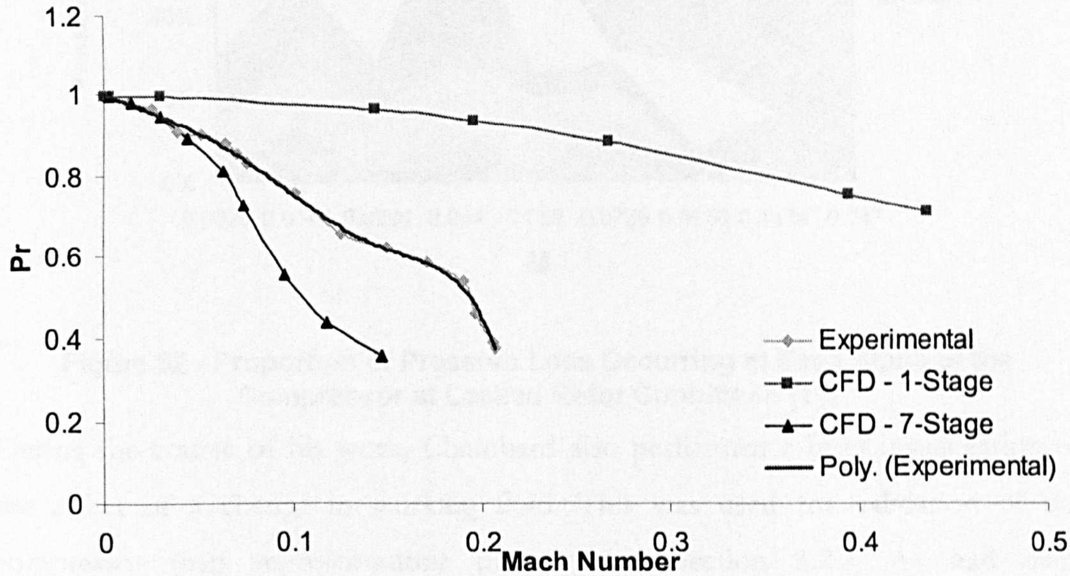

Figure 51 - Comparison of CFD Results for the Locked Rotor Analysis of a Compressor with the Results of Experiment [11]

The main difference seen between the CFD prediction and the experimental results was at high Mach number flows. Particularly, as the rear of the compressor moved towards choking, the results differed from those observed in practice. This is probably an effect of using the blade thickening method to simulate the reducing annulus area, a problem which it may be possible to tweak.

Another observation which was made was that the proportion of the pressure loss which comes from each stage does not follow a logical pattern, as shown in Figure 52. It was concluded that this was due to flow separation occurring. Where separation was encountered, the CFD simulation entered a cyclical mode, whereby, even though the residuals were small, the results were changing from one step of the iteration to the next. It is possible that this problem might be overcome through using an unsteady analysis.

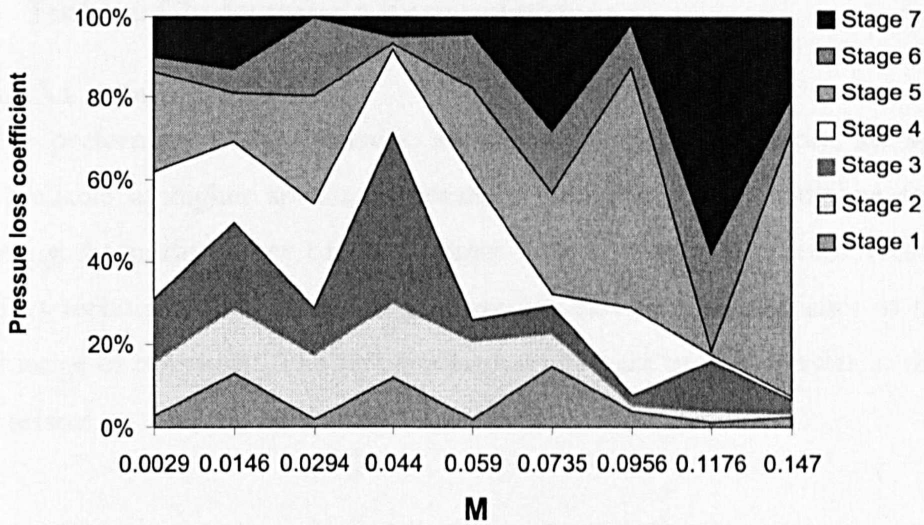

Figure 52 - Proportion of Pressure Loss Occurring at Each Stage of the Compressor at Locked Rotor Conditions [11]

During the course of his work, Chambard also performed a brief investigation of the effect of a change in working fluid. This was used for validation of the compressor map representations proposed in Section 3.2.2. As had been hypothesised, the viscosity had only a small effect on the flow, the majority of the effect being from the gas constant and the specific heat changes. This piece of work shows that the proposed method of representing the characteristic should give good predictions of performance, while the other methods will not give such accurate predictions.

CONCLUSIONS

It was demonstrated that CFD was capable of producing reasonable estimates of the locked rotor performance of a compressor. However, a few minor issues remain to be addressed. Therefore, the technique should be investigated further, as it shows potential in reducing the uncertainty of compressor characteristic extrapolation.

3.3.3 Turbine Characteristic Extrapolation

3.3.3.1 Introduction

Turbine performance characteristics, as with those of compressors, are generally only available at higher speeds of rotation. Therefore, ways must be found of predicting the performance of the turbines at low rotational speeds. This section describes techniques of extrapolating above-idle turbine characteristics to the low-speed range of operation. The techniques described are broadly similar to those for compressors.

3.3.3.2 Issues Encountered In Extrapolating Turbine Characteristics

Low Speed Turbine Operation

At low rotational speeds, turbine blades may stall if a high flow is passed over them. However, this will not normally be the case for steady-state windmilling.

Discontinuities in Variables

While it is possible for a turbine to act as a stirrer or as a compressor, this is not likely to be experienced in practice. Therefore, the issue of discontinuity in efficiency is not considered to be important.

Enthalpy Changes

Euler's equation (Eq. 6) describes the enthalpy change across the blade rows of a turbine as a function of flow angles. Unlike for a compressor, the air angles will normally be similar to the blade angles.

Losses

Turbine losses, like those for a compressor, can be described as being of three types: annulus, profile and secondary losses. Secondary losses are generally small unless the tip clearances are large. Annulus and profile losses behave as drags, the profile loss being dependent also on the angle of flow incidence.

Specific Heat Changes

Although specific heat changes are of great importance in the performance model, they are not an issue for the extrapolation process.

Reynolds Number Effects

Reynolds number effects are not normally an issue for turbines due to their high inlet temperature and pressure. However, they can be of significance while windmilling.

Variable Geometry

If a variable geometry turbine is being investigated, the extrapolation should be performed using fixed geometry maps and the composite characteristic then formed subsequently.

Cooling Flows

Turbine cooling flows influence the performance of the turbine. However, they do not present any problems specific to extrapolation.

Repeatability

Any technique for characteristic extrapolation should be repeatable. Should this not be the case, relative differences between the windmilling performances of different engine designs will not be possible.

3.3.3.3 Application of Extrapolation Techniques to Turbines

Introduction

The techniques used for compressor characteristic extrapolation in Section 3.3.2 are generally applicable to turbines also. This section therefore discusses how they can be applied, if at all.

YAN & MAI

Yan and Mai suggested the fitting of polynomial curves to stage characteristics to form the basis for the extrapolation. The instability seen through stacking characteristics defined using polynomials (Figure 47) is not as much of an issue for turbines as for compressors, as less stages are usually used and the turbine normally operates at nearer to the design point at off-design conditions. However, the issue that stage characteristics will not normally be available still holds true, although these could be estimated through characteristic destacking methods.

AGRAWAL & YUNIS

The approach of studying the performance on a single running line is appropriate for ground starting, but not for windmilling. The use of efficiency is not a limitation for turbines as it was for compressors.

CONVERSE & GIFFEN

In principle, the backbone method seems appropriate for turbines as well as for compressors. However, as seen in Figure 28, the pressure loss does not experience a minimum at all speeds and so it is not reasonable to apply the method.

RIEGLER ET AL.

Using the arguments of Riegler et al., the trivial point of zero flow and zero speed gives zero torque, zero enthalpy rise and zero ideal enthalpy rise. The zero speed line has zero enthalpy rise, but an accelerating torque and a negative ideal enthalpy rise. A graphical software, SmoothT, can then be used to perform the extrapolation as for compressors.

ZERO SPEED LOSS AND TORQUE CURVES

The generation of a locked rotor characteristic can aid the extrapolation by providing a lower bound and turning the problem into one of interpolation. The zero speed curves can be determined by experiment, computational analysis or correlation. Experiment is substantially easier for turbines than for compressors, as the turbine test rig is normally capable of performing tests with a locked rotor, although the instrumentation accuracy will not be as high as ideal.

ALTERNATIVE STAGE-BY-STAGE EXTRAPOLATION TECHNIQUE

Building upon the principle employed by Yan and Mai of looking at the stage characteristics for extrapolation, the basis of a stage-by-stage approach to extrapolating turbine characteristics has been developed as part of this project. This is explained in Section 3.3.4. The technique does, as for compressors, look very attractive.

GENERALISED CHARACTERISTICS

The use of generalised characteristics for sub-idle performance prediction is a reasonable tool for prediction during the early stages of engine development. However, better results will be found if a true extrapolation is used.

3.3.3.4 Comparison Between the Techniques

The Yan and Mai method is limited by the lack of availability of stage characteristics; the use of polynomial approximations does not present such severe problems for turbines as for compressors.

The method employed by Agrawal and Yunis of looking only at a running line is not appropriate for windmilling conditions.

The Converse and Giffen method is limited, for turbines, due to the lack of availability of data on the minimum pressure loss. A similar problem is encountered with compressors, though it is not as extreme as for turbines.

The use of an enthalpy rise, a technique used in all the methods, is good as it results from Euler's equation. However, specific torque is a better variable still as it is defined for locked rotor conditions.

Engineering judgement is required for most methods, leading to some variability between extrapolations of the same component by the same technique. The stage-by-stage method is considered superior in this respect. The use of a zero-speed characteristic, though difficult to generate, will also improve the repeatability.

3.3.3.5 Conclusion

The use of a zero-speed characteristic is recommended. This provides a lower bound to the extrapolation, thus making it more reliable.

Problems were encountered with most of the techniques described, though the stage-by-stage extrapolation technique described in Section 3.3.4 looks to be the most attractive method.

3.3.4 Stage-by-Stage Semi-Empirical Characteristic Extrapolation Technique

3.3.4.1 Introduction

As described in Section 3.3.2, one potential method of extrapolating a turbomachinery characteristic is the stage-by-stage semi-empirical method. This method is described in this section. The work of three SOCRATES exchange students is first presented, followed by details of how their work is incorporated into the overall method.

3.3.4.2 Method Overview

The deviation of the flow around a blade row and the losses which the flow experiences can be described by various empirical correlations. These are often employed to perform a simple analysis of a new design of a turbomachinery component.

Through the use of these correlations and Euler's equation, a turbomachinery characteristic can be assembled by varying the inlet conditions to the component.

The characteristic which results from such an analysis will probably reflect the component's real performance to a degree. However, it will probably not be sufficient to be reliable for use in a performance synthesis analysis.

The technique can potentially be modified to better reflect the off-design behaviour of the component. Given that off-design characteristics of components are normally readily available for above-idle operation of a gas turbine engine, even from the early stages of design, though becoming considerably more accurate throughout the design process, data are available to calibrate the empirical correlations against. Therefore, the correlations may be tweaked until a good match is obtained between the characteristic generated by a stage-by-stage analysis using the correlations and the known above-idle characteristic. Once the two characteristics match, it is logical to assume that the correlations may be used in regions where the component's performance is not already known.

Besides the fact that it should produce relatively reliable and repeatable results, the technique could be reasonably well automated through a software tool, allowing the analysis to be conducted by most engineers.

3.3.4.3 Stage-by-Stage Method for Compressor Characteristic Extrapolation (Part 1) [56]

The following work was performed by Joan Josep Moncholí, an exchange student at Cranfield University through the SOCRATES scheme and from la Polytecnica de Valencia, supervised by Prof. Pilidis and managed by the author of this thesis.

Moncholí investigated the feasibility of using an analysis of the deviations and losses across the individual stages of a compressor to determine the overall performance characteristic.

First, he took the design-point performance data for a compressor and produced a preliminary design of the blading for that compressor. As blading data will not be available for the early stages of development of an engine, but analysis of that engine's performance still needs to be carried out, this is an important first step.

Once an estimate of the blading had been produced, he then investigated the compressor's off-design behaviour. This was done by investigating the flow entering and leaving each blade row. Simple correlations were used to examine the flow deviation (Eq. 7), the profile loss (Eq. 8), the annulus loss (Eq. 9) and the secondary loss (Eq. 10), using the velocity diagram shown in Figure 53. It should be noted here that the deviation correlation assumes a constant deviation at off-design conditions. Using these correlations, the off-design performance of the compressor was mapped.

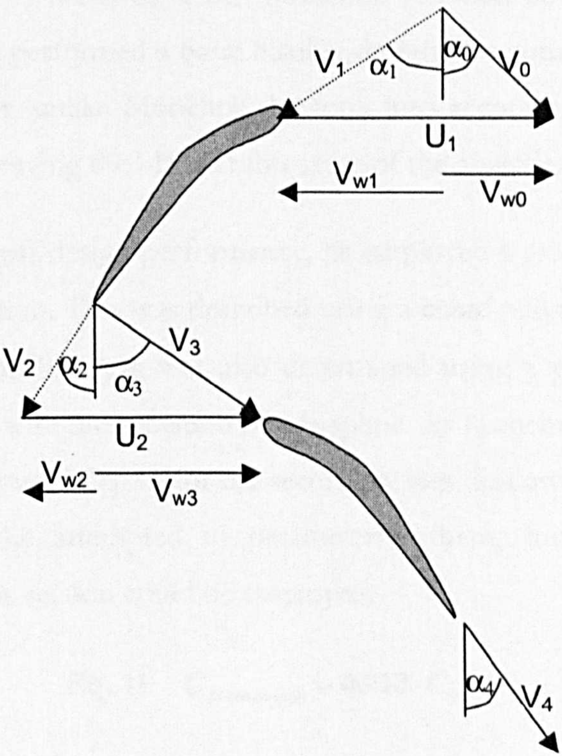

Figure 53 - Velocity Vectors and Angles Used

$$\text{Eq. 7} \qquad \tan(\alpha_2) = \tan(\beta_2) \cdot \left(1 - \frac{t_e}{t}\right)$$

$$\text{Eq. 8} \qquad C_{D,profile} = \frac{0.99}{\sigma} \cdot \cos^3(\alpha_m) \text{ where } \sigma = \frac{c}{s}$$

$$\text{Eq. 9} \qquad C_{D,annulus} = 0.02 \cdot \frac{s}{h}$$

$$\text{Eq. 10} \qquad C_{D,secondary} = 0.29 \cdot \frac{C_T}{h} \cdot C_L^{3/2}$$

3.3.4.4 Stage-by-Stage Method for Compressor Characteristic Extrapolation (Part 2) [24]

The following work was performed by David Gattini, an exchange student at Cranfield University through the SOCRATES scheme and from Perugia University, supervised by Prof. Pilidis and managed by the author of this thesis.

Gattini built upon the methods which Moncholí [56] had developed. Adopting a similar approach, he performed a basic blading design of a compressor from design point data. However, unlike Moncholí, he took into account the variation of the deviation of the air leaving the blade at this stage of the analysis.

In investigating the off-design performance, he employed a graphical correlation to determine the deviation. This was described using a cubic polynomial and is shown in Figure 54. The profile drag was also determined using a graphical correlation, shown in Figure 55, with interpolation by a b-spline. As Moncholí, he used Eq. 9 for the annulus loss, but used Eq. 11 for the secondary loss. Following the employing of these correlations, he attempted to parameterise them, such that the method described later in this section could be employed.

$$\textbf{Eq. 11} \quad C_{D,secondary} = 0.018 \cdot C_L$$

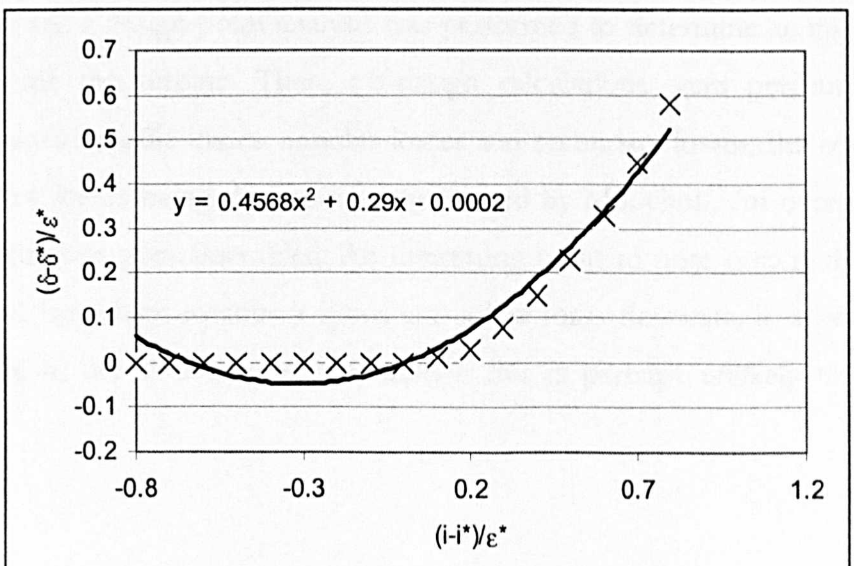

Figure 54 - Deviation Correlation Used for a Compressor by Gattini [24]

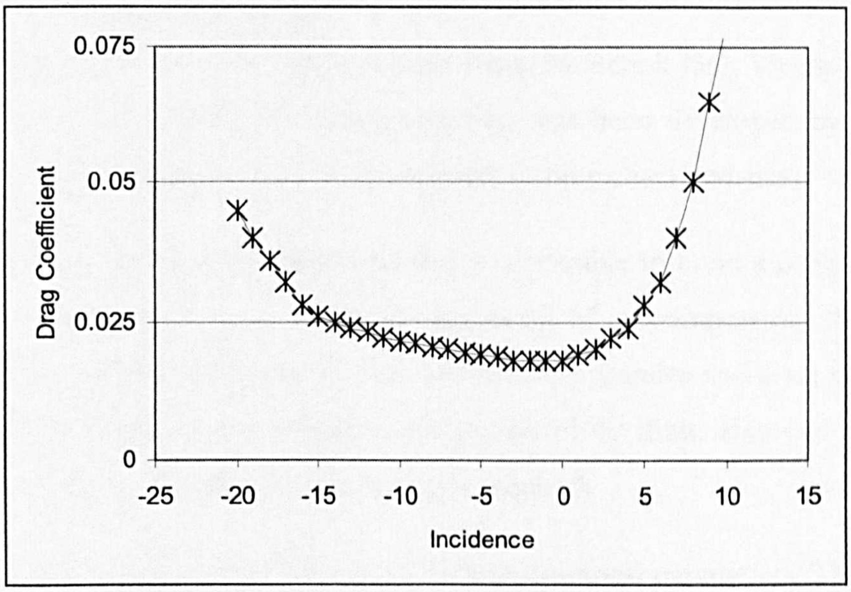

Figure 55 - Profile Drag Correlation Used for a Compressor by Gattini [24]

3.3.4.5 Stage-by-Stage Method for Turbine Characteristic Extrapolation[14]

The following work was performed by Vicent Chulvi, an exchange student at Cranfield University through the SOCRATES scheme and from la Polytecnica de Valencia, supervised by Prof. Pilidis and managed by the author of this thesis.

Chulvi applied similar techniques to those of Moncholí [56] and Gattini [24] to turbines. First, a design-point analysis was performed to determine an approximate geometry for the turbine. Then, off-design calculations were performed using correlations for profile losses, annulus losses and secondary losses, the correlations used for the losses being the same as those used by Moncholí. An overall turbine characteristic was then assembled. An interesting point to note here is that Chulvi shows that, for a high rotational speed and a low mass flow rate, it is possible for the turbine to act as a compressor, though this is perhaps unlikely to occur in practice.

3.3.4.6 Continued Work

Building on the work of the students Joan Josep Moncholí [56], Vicent Chulvi [14] and David Gattini [24], the following technique has been developed by the author of this thesis for extrapolating compressor and turbine characteristics.

The three students have demonstrated that it is possible to build a compressor or a turbine characteristic using the basic geometry of a component, the physical equations governing compressible flow and thermodynamics and a set of empirical relationships. However, the characteristic produced by these methods will not be perfectly accurate. Therefore, some tuning is required.

The correlations used in Eq. 7 to Eq. 11 include empirical parameters. The graphical correlation of Gattini replaces Eq. 9. Therefore, the equations can be rewritten in Eq. 12 and Eq. 13. The charts used by Gattini can be scaled by the factors shown in Figure 56 and Figure 57. The resultant relationships contain eight parameters which may be altered. This is repeated for each stage, with the parameters being independent of one another. Thus, the number of parameters used is eight multiplied by the number of stages.

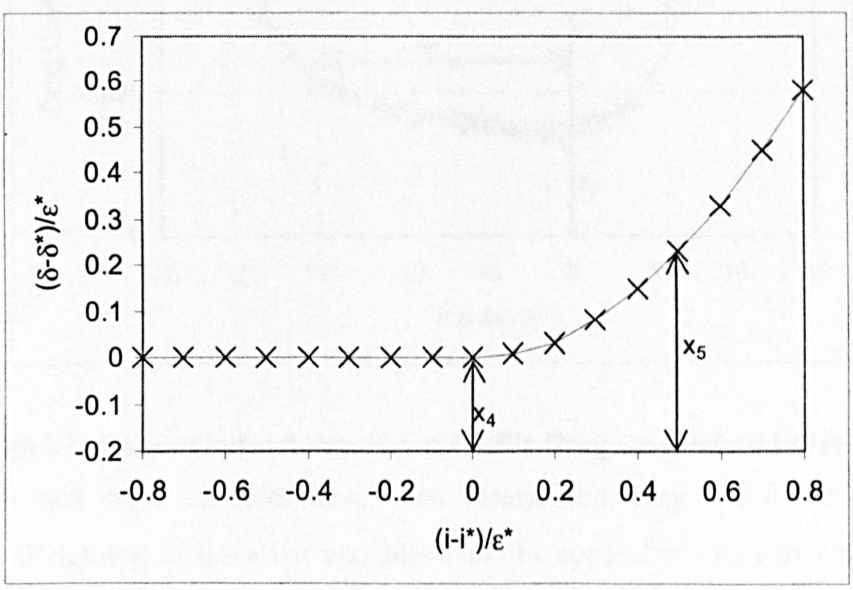

Figure 56 - Parameterised Version of Deviation Correlation

$$\text{Eq. 12} \quad C_{D,annulus} = \frac{x_1}{\sigma} \cdot \cos^3(\alpha_m)$$

$$\text{Eq. 13} \quad C_{D,secondary} = x_2 \cdot \frac{C_T}{h} \cdot C_L^{\,x_3}$$

If estimates are made for each of the parameters, a resultant characteristic may be built. This can then be compared to the known above-idle characteristic. The fit of the two characteristics can be determined by using two root-mean-square coefficients. These can be found by taking a large number of points along known speed lines and looking at the difference between the known characteristic and the predicted map. This comparison should be conducted both for the specific torque and the pressure loss and must be made at a far larger number of points than the number of parameters to be determined.

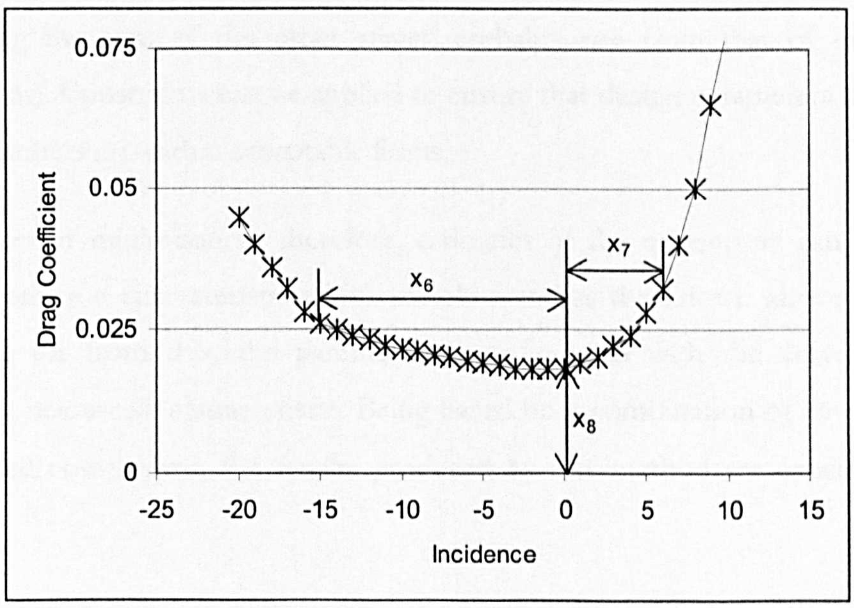

Figure 57 - Parameterised Version of Profile Drag Coefficient Correlation

Once the two error variables have been determined, they should be combined together. Weighting of the error variables must be applied in order that the specific torque fit does not swamp that of the pressure loss or vice versa, although this weighting has not been determined in this project.

The result of this is a set of estimates of the parameters and a resultant error function. Using a large number of such estimates, a minimum of the error function may be determined by an error minimisation technique. If necessary, constraints may need to be applied. A number of techniques exist for the solution of such a problem, although the experience at Cranfield is that genetic algorithms are well-suited to such an error minimisation problem.

Should the number of parameters prove to be too many, some of them may be removed through removing their independence from one another. Conversely, if the geometry of a component is not known, the optimisation should include the blading. This will of course be the case during the preliminary design phase of an engine. Here, the main factor is the proportion of the design point enthalpy rise given to each stage. This can be set up easily as an enthalpy rise parameter for each stage (except the last stage, for which the enthalpy rise can be calculated from subtracting the sum of the other stages' enthalpy rise from that of the overall component). Constraints can be applied to ensure that design parameters such as de Haller numbers are within acceptable limits.

Through error minimisation, therefore, estimates of the parameters can be found which produce a characteristic which closely matches the known above-idle map. Following on from this, the parameters may be used with the correlations to produce a below-idle characteristic. Being based on a combination of sound theory and trusted correlations, the results produced by this method are expected to be sound.

3.4 VERIFICATION OF METHODS

3.4.1 Introduction

During the course of this work, methods have been developed to both represent and create sub-idle performance characteristics for turbomachinery components. In this section, methods for further validating the methods proposed are presented.

3.4.2 Characteristic representations

Comparisons have been performed between the methods of representation methods. While the method proposed is based on sound theory, its validation is somewhat difficult. In order to be able to thoroughly validate that the representation is indeed the best, comparison must then ideally be made with experimental results. However, such an experiment must be capable of providing to the component values of pressure, temperature, composition and mass flow which vary independently of one another and also have very accurate instrumentation to detect the small changes which would be expected. In practice, it is unlikely that such an experiment would be devised for a real compressor or turbine.

One experimental approach which could be adopted would be with the use of alternative fluids. In passing a different gas through a component, the values of gas constant and specific heat would be changed. Therefore, it would be possible to investigate whether the proposed methods of representation held given the change of fluid; if they did, it would demonstrate that the method would hold for the relatively small changes in gas constant and specific heats encountered in windmilling and relight operation. Nevertheless, the large quantities of gas which would be required to perform such a test on a real compressor or turbine make such an experiment impractical. Furthermore, it would not be appropriate to conduct the test on a geometry much simpler. Firstly, it is necessary to have the relative movement of rotors and stators. This effectively requires rotation. Then, the flow must be over blades. Finally, the annulus area must change throughout the length of the component. These three requirements effectively mean that the test must be conducted on a real component.

An alternative to experiment is computational analysis, through Computational Fluid Dynamics. This has already been performed for the locked rotor case by Romain Chambard [11], an M.Sc. student working on this project. In his work, he showed that the representation method proposed worked for changes in the gas constant and specific heats. Furthermore, he showed that the effect of viscosity changes was small, the main area of concern in the methods proposed. However, only the locked rotor case was studied, and the computational method is of limited accuracy when flow separation is encountered. Nevertheless, relative comparisons with different fluids should produce reliable effects.

3.4.3 Characteristic extrapolations

3.4.3.1 Introduction

In order to verify that the extrapolation techniques provide sufficiently accurate results, one or more tests should be conducted. A variety of possibilities for the design of this test are presented here and their usefulness reviewed.

3.4.3.2 Component to Test

CHOICE OF TURBOMACHINERY COMPONENT TYPE

As compressors are expected to present more problems than turbines in respect of the extrapolation of characteristics, it is reasonable to assume that an extrapolation technique which produces valid results for compressor characteristics will do similarly or better for turbines. Therefore, testing should be performed on a compressor.

A multi-stage compressor is necessary in order to effectively study the effect of the reducing annulus area. Therefore, a test on a fan will not be suitable for the prediction of the operation of a core compressor.

The drag of an engine is generally produced by the fan. However, for engines which do not have booster stages on the low-pressure shaft of the engine, the core compressors control most other aspects of windmilling performance. This particularly includes the problem areas of power offtake availability and compressor efficiency for pull-away.

The operation of a fan during windmilling is somewhat different in that the two-dimensional flow is likely to be the dominant factor in determining its performance.

Therefore, if tests were to be conducted on the two-dimensional nature of fan operation, it would perhaps prove informative to study a number of test points in the sub-idle range with a high bypass ratio.

CHOICE OF PRODUCTION COMPONENT OR EXPERIMENTAL DESIGN

A number of experimental compressors exist in various laboratories, including at Cranfield. These could potentially hold some advantages over production compressors. This section therefore discusses the relative merits of using a production or an experimental compressor for the purposes of this type of study.

In order that a test not be specific to one compressor, it is preferable that it be conducted with various geometries. This can be most effectively achieved with an experimental compressor, in which the blades may be changed for those of different profiles and where the angles of the blades may be altered. However, the effect of alternative geometries can also be tested on a variable geometry production compressor. A test on such a component will not be suitable for predicting the effect of a change of blade profile or size. Ideally, if a production compressor is to be tested, it should have a number of blade rows with variable geometry and be capable of being modified to allow each row to turn independently in order to allow a large number of configurations of geometry. A further change of geometry which is possible with a production compressor is the opening of bleed valves to change the flow rate of air over the later stages.

An important consideration is that instrumentation must be relatively easy to fit and should not disturb the flow significantly. If measurements are to be taken in an attempt to quantify the proportion of the different loss mechanisms, these measurements are likely to include a radial rake of instruments. Therefore, a compressor with a large annulus area would be preferable. However, as air must be blown through the compressor, a reduction in annulus area will produce a reduction in air requirement and, consequently, in cost. Clearly, a compromise must be attained between these differing constraints.

Testing on a production compressor will allow a comparison to be made between the rig test and in-engine tests. This will be useful for the calibration of the rig. As the Trent family of engines make up a large proportion of Rolls-Royce's sales, one of this family seems an appropriate choice. As the intermediate pressure compressor has stability bleeds, inlet guide vanes and two rows of variable stator vanes, this seems to be more suitable for testing than a high pressure compressor.

For the above reasons, it is recommended that the tests be conducted on a production compressor which has been used, or will be used, in an engine which has undergone windmilling testing. The choice of the size of compressor has not been determined as there are conflicting requirements for measurement precision and operating cost. Therefore, the choice of engine has not been made.

It is therefore recommended that the tests be conducted on a Trent intermediate pressure compressor, with the choice of engine to be decided by the practicalities of instrumentation fitting, test running costs and availability of good in-engine windmilling data.

3.4.3.3 Testing Requirements

The tests should be ideally used to verify the following:

- Sub-idle compressor characteristic
- Zero-speed torque and loss curves
- Proportion of pressure loss and torque occurring across each stage
- Proportion of pressure loss associated with each loss mechanism
- Changing geometry

The main aim of the test would be to verify that the extrapolation methods produce reasonable estimates of the whole sub-idle characteristic. If a zero-speed characteristic has been generated for the extrapolation by CFD or other means, it is preferable to measure the performance away from the locked rotor case in order to verify the interpolation; if a zero-speed line has not been used, testing at only very low speeds will suffice, as this then represents the most extreme case for the extrapolation. Testing only at conditions where the compressor is absorbing power from the airstream (and therefore acting as a turbine) will prove to be useful. However, particularly to investigate the region of interpolation between a locked rotor condition and above-idle operation, it would be beneficial to study the component when it is being motored.

In the testing, the locked rotor case should be studied in some detail. This represents the extreme of the extrapolation, and is also the case against which a CFD study should be compared. The locked rotor tests need to be conducted a number of times, with the rotor locked in different positions in order to avoid the clocking effects which were found by Chambard [11] during the course of this project. As measurements of enthalpy changes will not show the torque acting on the compressor, with no work being performed, this must be measured by a torque meter on the spool.

Some inter-stage measurements would demonstrate the proportion of the losses and enthalpy changes occurring across each stage. It should be noted that this is not necessary at every stage, although the first and last stages should definitely be studied. As some of the blades may be stalled, a number of measurements should be made circumferentially.

The measurement of which types of losses (profile, annulus or secondary) are the dominant mechanisms and how they vary over the sub-idle operating range would prove to be valuable in improving understanding of the operation of the component. However, in order to perform such measurements, extensive instrumentation would be required that is probably impractical to achieve. Therefore, such studies are perhaps better carried out by computational means.

With a variable geometry compressor, some understanding of the influence of profile losses and how they vary can be gained. Furthermore, both through altering the inlet flow area and through the use of bleed valves, the influence of the reducing annulus area can be checked.

3.4.3.4 Basic Rig Types

In order to check the accuracy of an extrapolated component characteristic, it is necessary to provide a number of test points against which to compare the predicted and observed values. Therefore, the component must be tested throughout the range for which the map has been extrapolated. In the case of a compressor, this means that the testing should ideally be conducted with the component acting in each of its three modes: compressor, stirrer and turbine. To achieve this, a compressor test rig will not suffice.

Compressor test rigs, such as shown in Figure 77, are capable of controlling the flow well when the pressure ratio between compressor outlet and ambient is high enough for the throttle to choke. Therefore, without modification, they are not capable of testing the performance of a compressor with the low pressure ratios in which we are interested.

Turbine test rigs, as shown in Figure 78, are more suitable for testing compressor windmilling operation than are compressor test rigs. However, the compressor may only be tested under conditions where it acts as a turbine, as the rig will not normally be designed to motor the component. This is not an excessive restriction, as we are generally more interested in the compressor operation in this range for validation purposes; this is the more extreme area for extrapolation, so good fitting here will suggest good fitting at higher speeds and positive enthalpy rises.

A further alternative is provided by altitude test facilities or by wind tunnels. A compressor can be fitted to an altitude test facility relatively easily. As only a core compressor is to be tested, a small facility will suffice. No chiller plant is required. Such a layout is shown in Figure 58. The compressor may be fitted to a wind tunnel in a similar manner, as in Figure 59. It should be noted that the test section need only be large enough to physically fit the compressor, as we are interested in internal flows, not external.

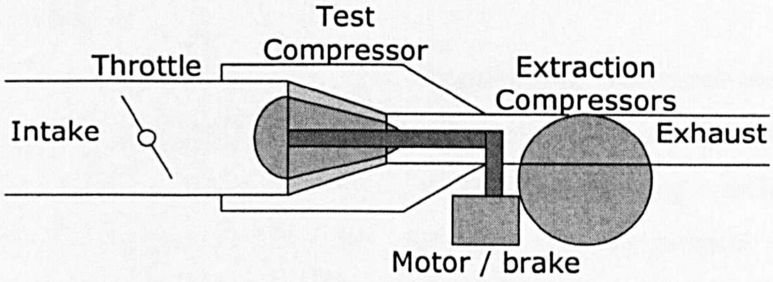

Figure 58 - Testing of a Compressor in an Altitude Test Facility

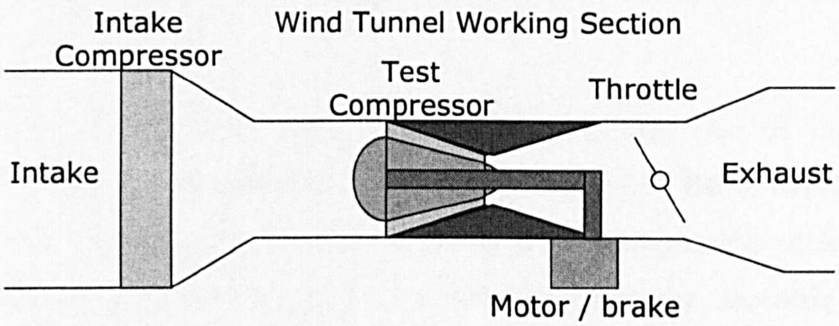

Figure 59 - Testing of a Compressor in a Wind Tunnel

In summary, a compressor test rig is not suitable for testing the sub-idle performance. A turbine test rig could be used, though it would need to be fitted with a motor suitable for powering the compressor and appropriate torque measurement. An altitude test facility or a wind tunnel may provide the best environment for testing.

3.4.3.5 Resultant Rig Design

The previous subsections have described some aspects of the design of an experimental rig for testing the extrapolation of turbomachinery characteristics. Arguments have been presented for the use of an intermediate pressure Trent compressor. This should be mounted in a turbine test facility, an altitude test facility or a wind tunnel. It should be heavily instrumented and a series of tests should be conducted around the sub-idle operating range, particularly at locked rotor conditions.

3.4.4 Summary

Some validation of the method for characteristic representation has been performed. During the development of a locked rotor computational model of a gas turbine compressor, an analysis was made of the effect of using a different working fluid. This demonstrated that the method of compressor characteristic representation was appropriate. However, ideally further analysis should be conducted. For reasons of practicality, this should be computational rather than experimental.

With regard to the extrapolation of characteristics, only one of the methods presented will give results which are reasonably repeatable. Being based on sound theory, this is the method of choice. However, in order to provide validation of the extrapolations, experiments can be conducted to test the methods proposed. Possible configurations for these experiments have been highlighted.

3.5 SUMMARY

In this chapter, methods for representing both compressor and turbine characteristics in a gas turbine performance model have been presented and the problem of how to generate the sub-idle characteristics has been addressed. Means of validating the methods have been presented and have been applied where possible.

In order to represent the performance of both compressors and turbines in a performance simulation, the use of non-dimensional speed, beta, non-dimensional flow rate, specific torque and modified pressure loss is recommended.

For the extrapolation of turbomachinery techniques, the semi-empirical stage-by-stage method should be developed further. This should then be coupled to a locked rotor computational model to provide reproducible and reliable results.

4 Performance Model

4.1 INTRODUCTION

In order to test the effect of altering characteristic representations and extrapolation techniques and to test the robustness of the solver methods, it was necessary to build a simple gas turbine performance code. This chapter describes the philosophy behind and the workings of the simulation code which was produced and assesses its robustness.

4.2 PRELIMINARY WORK [35]

The work presented in this section was performed by Sog-Kyun Kim [35], supervised by Dr. Pilidis and guided by the author of this thesis. The work was then published as a paper [36].

Kim built a transient gas turbine performance model using the SIMULINK extension to MATLAB and based on the inter-component volume method [69] of analysis. Using this performance model, he attempted to simulate the transient processes of spool-down and pull-away. He also attempted a similar analysis using a gas turbine performance program written in FORTRAN and based on the continuity of mass flow method of transient simulation.

Turbomachinery characteristics were stored in the form of pressure ratio and efficiency against speed and flow. The compressor map used was one which had been extrapolated to the low speed region. The turbomachinery characteristics were scaled to the appropriate values for the engine being modelled.

Problems were encountered in performing steady-state analysis at low values of fuel flow in both models. However, the SIMULINK model was substantially more robust in this respect. In the transient simulations, the SIMULINK model was capable of producing a final, stabilised value of speed following a fuel-chop.

However, the FORTRAN model was not. From this stabilised value of speed, the SIMULINK model was capable of modelling the pull-away acceleration of the engine.

This work demonstrates that standard methods of gas turbine performance simulation may not work reliably for windmilling and relight modelling. However, the more robust solver provided with SIMULINK was capable of performing the flow matching for a transient analysis reasonably reliably.

4.3 Model Details

The engine which was modelled in this project was a single-spool turbojet engine. As the only effects being studied were the influences of turbomachinery characteristics, the effect of power offtakes and the working of the solver, it was considered to be unnecessary to model many of the effects which should be modelled in a performance simulation of a real gas turbine engine. Therefore, the intake, combustor and nozzle were considered to be free of losses. While this is not really the case, such a simplification makes little difference to the effects which were being studied.

The project was specified to be for steady-state calculations and thus transient effects are neglected.

The model was based upon a Rolls-Royce Viper 11 engine in terms of design point sizing. However, as actual component characteristics were difficult to obtain, generic characteristics have been used throughout.

The basic design point sizing was taken from Jane's [32] and from the reports of Chulvi [14], Gattini [24] and Moncholí [56]. This is shown in Table 7.

Table 7 - Design Point Data Used for Turbojet In Study

Mass flow rate / kg s^{-1}	20
Compressor pressure ratio	4.8
Compressor polytropic efficiency	0.9
Compressor inlet total pressure / Pa	101325
Compressor inlet total temperature / K	288
Compressor mean inlet axial velocity / m s^{-1}	200
Compressor mean inlet blade speed / m s^{-1}	275
Compressor mean inlet diameter / m	0.362
Compressor inlet area / m^2	0.0986
Turbine inlet total temperature / K	1500
Turbine inlet Mach number	0.3

4.4 SOLVER STRATEGY

4.4.1 Introduction

This section describes the choices of variables to be used in the simulation, the solver to be used, the strategy which was employed to ensure convergence of the solver and the overall resultant model.

4.4.2 Handles

In order to run a gas turbine performance simulation, it is necessary to supply to the simulation certain variables which describe the environment in which the engine is operating. As the focus of this thesis is on high altitude windmilling, the free stream pressure, temperature, humidity level and ram ratio are required. In relight studies, the required output is the relight envelope of the engine. Similarly, for windmilling drag studies, the interest is on the variation of drag around the flight envelope. In this study, the effect of variation of air conditions away from ISA standard can be neglected and thus altitude, flight speed and relative humidity were chosen to describe the operating environment.

Besides having to describe the external operating environment of the engine, the performance simulation must also be supplied with information on the operational settings. These include whether certain devices are operational and information on the power setting of the engine.

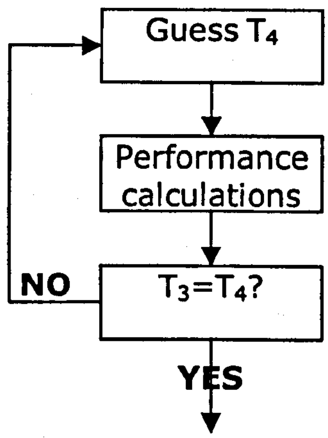

Figure 60 - Iteration Required for Windmilling Modelling if TET Is Used as a Handle

In many performance simulations, the turbine entry temperature is used as a handle, as this ensures that the components all operate at sensible conditions throughout the iteration, even when the solver attempts to use inappropriate values of the guess variables. This is in contrast to the use of the fuel flow rate, where, if the solver guesses a compressor operating point which produces a very low air flow rate, the resultant turbine entry temperature will be very high. However, the use of turbine entry temperature is inappropriate for this simulation, as windmilling occurs at zero fuel flow and thus an extra iteration would be required to find the turbine entry temperature which was produced by the zero fuel flow condition (Figure 60). Therefore, the usage of both fuel flow rate and turbine entry temperature present problems if the model is to be used for both windmilling and powered simulations. An alternative power setting handle which avoids both these problems, however, is the combustor fuel-air ratio. However, as with using turbine entry temperature as a handle, it is possible that two different operating points could be found which have the same combustor fuel-air ratio. Therefore, it is recommended that the mass flow rate of fuel be used.

The settings for determining whether the hydraulic pumps are pressurised and whether the starter motor is being used are implemented as simple switches.

Fuel and oil temperatures are set during the main simulation, as these are calculated from the total temperature of the free stream air.

4.4.3 Requirements for Guesses

In a performance simulation, it is necessary to guess the operating point of the engine and to then determine whether this guess satisfies the necessary constraints [21] (See section 2.3.2).

The choice of variables to use for these guesses can influence the ease of programming and the speed and stability of the iteration.

For ease of programming, it is sensible to use the variables which define the operating points of the components. Thus, the compressor and turbine beta variables are employed. These are then used in conjunction with the compressor non-dimensional speed.

In order to ensure stability of the solver, the behaviour of the engine should be smooth over the allowable range of variables. Guesses of compressor and turbine betas between zero and one, and guesses of non-dimensional speed over roughly the same range, all produce valid compressor operating points and, if the turbine entry temperatures remain over a sensible range, valid turbine operating points also.

4.4.4 Requirements for Errors

If a multi-dimensional solver is used, all error variables should ideally have similar magnitudes, as the solver will usually attempt to minimise a root-mean-square of the errors. The error variables must also be linearly independent of each other and at least one must be affected by a change in any of the guess variables. The error surface must be smooth and continuous. Ideally, there should be only one solution.

The magnitude of the errors should not change with rotational speed and a zero rotational speed solution should be possible.

Especially during windmilling, poor choices of turbomachinery operating points, particularly for the compressor, can produce a sub-atmospheric nozzle. This should not present problems for the iteration, but rather provide a large error of the correct sign which increases as the nozzle pressure becomes lower.

For the above reasons, it was decided to use full non-dimensional variables for all the errors.

Flow matching was chosen for two of the error variables. As the calculations for the engine progress more logically when done in the direction of the airflow, flow matching at combustor outlet and at turbine outlet were chosen.

The third and final error variable is regarding the balance between the powers or torques acting on the engine spool. In order to ensure that the error variable was of consistent magnitude throughout the range of rotational speeds, and that it was defined for the locked rotor case, non-dimensional torque was chosen. This variable is shown in Eq. 14 and is essentially the product of specific torque and non-dimensional flow rate. Note that the torques must all be non-dimensionalised against the same diameter, temperature, pressure, area, gas constant, specific heat and gamma, with only the enthalpy rise and mass flow rate changing between the components. The power offtakes are included in a similar fashion.

Eq. 14 $$\tau_{nd} = \frac{\Delta H}{N \cdot \pi \cdot d_2 \cdot \sqrt{\gamma_2 \cdot R_2 \cdot T_2}} \cdot \frac{W \cdot \sqrt{R_2 \cdot T_2}}{A_2 \cdot P_2 \cdot \sqrt{\gamma_2}} = \frac{\Delta H \cdot W}{\gamma_2 \cdot N \cdot \pi \cdot d_2 \cdot A_2 \cdot P_2}$$

$$= \frac{\tau_{spec} \cdot Q}{\gamma_2/\gamma \cdot d_2/d \cdot A_2/A \cdot P_2/P}$$

4.4.5 Nozzle Operation

Nozzle calculations are performed by first investigating which mode the nozzle is operating in. That is, whether it is choked, unchoked or sub-atmospheric. If the nozzle is choked, the entry conditions used in the flow matching are the turbine exit temperature and pressure and the choked nozzle flow rate. If the nozzle is unchoked, the nozzle entry conditions used are the turbine exit temperature and pressure and the flow rate resultant from the pressure ratio across the nozzle. If the nozzle is sub-atmospheric, the nozzle calculations are performed as though the nozzle were operating in the opposite direction. That is, a set of calculations are performed on the nozzle in the reverse direction, with the entry pressure equal to the atmospheric pressure and the inlet temperature equal to the turbine exit temperature, and then the resultant mass flow rate is then calculated. Finally, the turbine exit temperature and pressure and the negative of the mass flow rate just calculated are used as the inlet conditions to the nozzle. This produces a nozzle characteristic as shown in Figure 61. As required, this is smooth and continuous and affects the flow matching error in a sensible fashion.

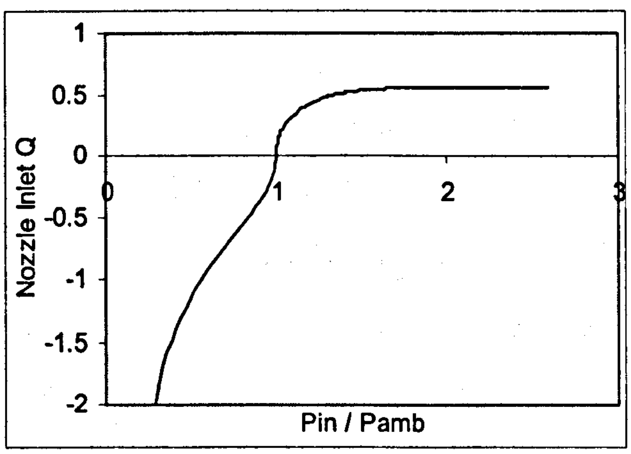

Figure 61 - Nozzle Characteristic Resultant from Solution Scheme

4.4.6 Solver

A number of solvers exist which have potential for use in gas turbine performance simulation. These include many Newton-Raphson solvers, bisection methods and genetic algorithms. As this work was conducted for Rolls-Royce, it was decided that their in-house solver, as used in RRAP, should be employed. This is a multi-variable Newton-Raphson solver with the facility for nesting of iterations and contains fall-back methods should the Newton-Raphson iteration encounter problems.

Initially, a simultaneous iteration on all three variables was attempted. While this performed reasonably well for even quite low power settings, however, it suffered from convergence problems as windmilling conditions were approached. This was due to the problem of there being potentially more than one solution which satisfies the necessary balances. Therefore, the solution scheme was altered to nest the iteration, with flow matching and betas forming an internal two-dimensional iteration and the spool speed and torque balance forming an outer iteration. The outer iteration was of the form of a line search to find the occurrence of torque balance with the highest rotational speed as this is usually the solution of most interest. The solution was then found by the bisection method.

The iteration scheme resultant from this will also permit simpler modification for transient analysis than a full three variable matrix iteration, as the outer loop may be replaced with a transient model.

4.4.7 Resultant Scheme

The station numbering scheme used in the calculation is shown in Figure 62.

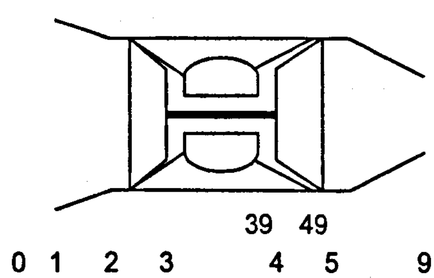

Figure 62 - Station Numbering Scheme Used for Single Spool Turbojet Calculations

The solution scheme which results from the above arguments is shown in Figure 63.

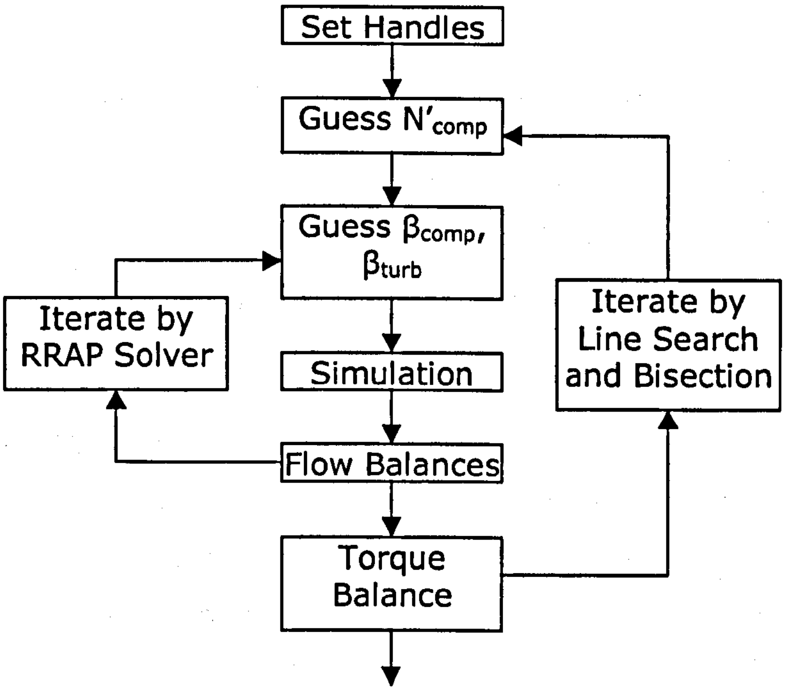

Figure 63 - Solution Scheme Used for Modelling a Single Spool Turbojet Engine

The variables used in the solution scheme are summarised below.

4.4.7.1 Handles

- Flight altitude
- Flight speed
- Relative humidity
- Combustor fuel flow rate
- Operation of hydraulic pumps
- Operation of starter systems

4.4.7.2 Guesses

- Compressor non-dimensional speed $\dfrac{N \cdot \pi \cdot d_2}{\sqrt{\gamma_2 \cdot R_2 \cdot T_2}}$
- Compressor beta
- Turbine beta

4.4.7.3 Errors

- Torque balance $\dfrac{\Delta H_{23} \cdot W_2 + \Delta H_{45} \cdot W_4 + Power_{offtakes}}{\gamma_2 \cdot N \cdot \pi \cdot d_2 \cdot A_2 \cdot P_2} = 0$
- Combustor exit flow balance $\dfrac{W_{39} \cdot \sqrt{R_{39} \cdot T_{39}}}{A_{39} \cdot P_{39} \cdot \sqrt{\gamma_{39}}} = \dfrac{W_4 \cdot \sqrt{R_4 \cdot T_4}}{A_4 \cdot P_4 \cdot \sqrt{\gamma_4}}$
- Turbine exit flow balance $\dfrac{W_{49} \cdot \sqrt{R_{49} \cdot T_{49}}}{A_{49} \cdot P_{49} \cdot \sqrt{\gamma_{49}}} = \dfrac{W_5 \cdot \sqrt{R_5 \cdot T_5}}{A_5 \cdot P_5 \cdot \sqrt{\gamma_5}}$

4.4.8 Extension to Multi-Spool Engines

Through the use of nesting in the iteration, multi-spool engines can be modelled without greatly increasing the complexity of the problem. This is because the spools can be considered almost in isolation, from the view of the solver scheme. This is in contrast to a full matrix iteration, where all variables are iterated simultaneously, and the problem thus becomes very large when the number of spools increases. Therefore, it is expected that a two-spool turbojet could be modelled as shown in Figure 64.

Due to the constraint of time, the extension of the scheme to multi-spool engines has not been pursued; this is an area which should be pursued by future researchers.

4.5 COMPONENT REPRESENTATION

4.5.1 Interchangeability of Map Representations

One of the reasons for building the single spool simulation was to test the effect which the choice of representation has on the results. To this end, the turbomachinery characteristics were stored in all suitable representation schemes within the program. When performing the simulation, the appropriate methods of compressor and turbine calculations were used.

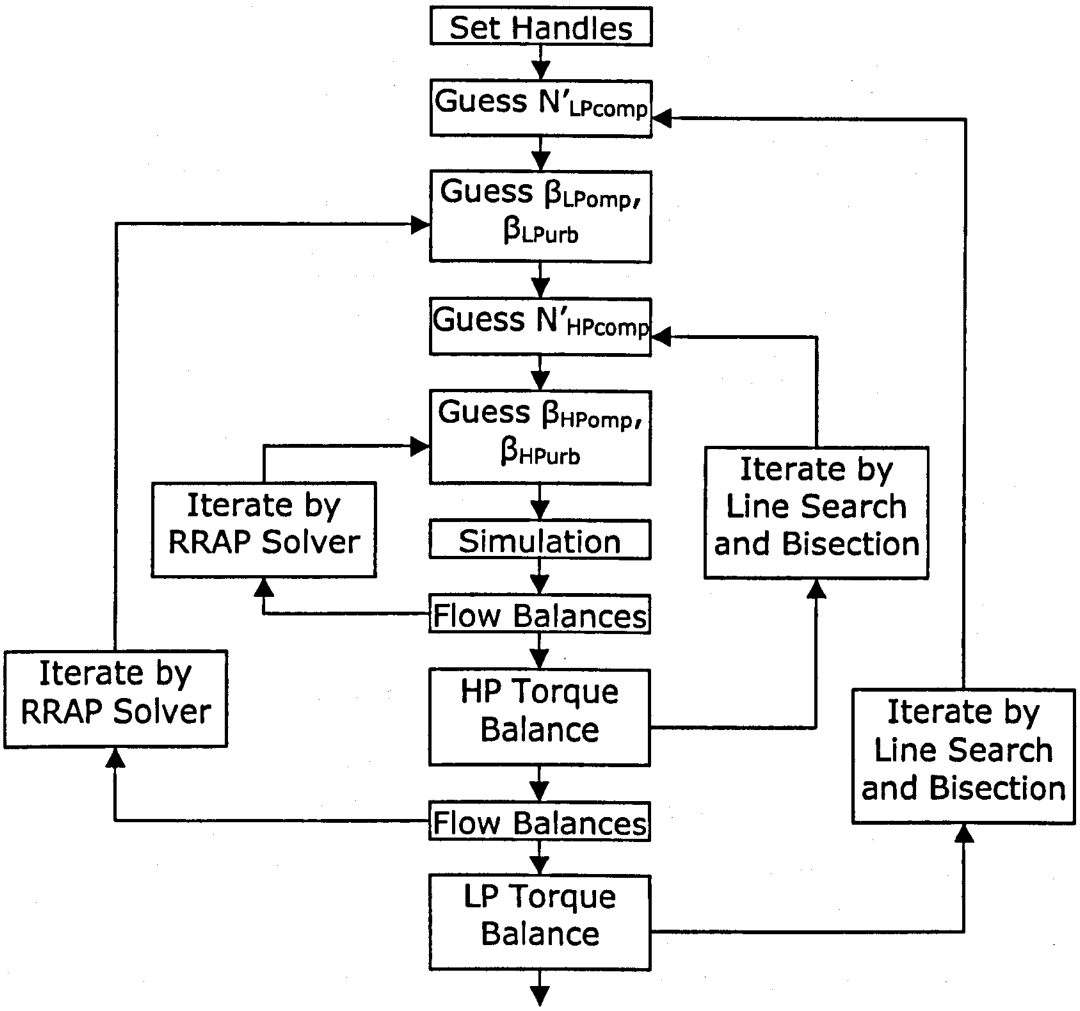

Figure 64 - Extension of Solver Scheme to a Two-Spool Turbojet

Tests were conducted using the representation methods shown in Table 6 (see Page 72).

The various representations were stored in the gas turbine performance program using a user-defined type to describe the characteristic. Thus, the following type was used to define a turbine, a compressor characteristic being held in the same format:

```
TYPE, PUBLIC :: TURBMAP !Turbine map data
  REAL*8 D
  INTEGER*4 NNRT
  INTEGER*4 NBETA
  REAL*8    NRT(20)       !N/rootT
  REAL*8    NCORR(20)     !N'
  REAL*8    BETA(20)      !Turbine beta
  REAL*8    WRTP(20,20)   !WrootT/P
  REAL*8    PRAT(20,20)   !Expansion ratio
```

```
      REAL*8     EFF(20,20)        !Turbine efficiency
      REAL*8     DHT(20,20)        !Array of values of DeltaH upon T
      REAL*8     DHTID(20,20)      !Array of values of DelatH id upon T
      REAL*8     PLOSS(20,20)      !Array of values of pressure loss
      REAL*8     MODPLOSS(20,20)   !Array of values of mod pressure loss
      REAL*8     TSPEC(20,20)      !Array of values of specific torque
      REAL*8     Q(20,20)          !Array of values WrootRT/AProotGamma
      REAL*8     DHN2(20,20)       !Array of values of DeltaH upon N**2
      REAL*8     DHIDN2(20,20)     !Array of values of DeltaHid on N**2
      REAL*8     WTNP(20,20)       !Array of values of WT/NP
    END TYPE
```

4.5.2 Generation of Scaled and Transformed Characteristics

To produce the characteristics, generic characteristics were scaled at design point on the variables N', Q, ($P_{out}/P_{in} - 1$) and efficiency. After the scaling, they were then converted into other forms using the values of gas constants and specific heats found at design point.

The process of scaling and transforming the characteristics was produced with a tool developed during this project. The basic, unscaled characteristic is copied into a Microsoft Excel spreadsheet. A design point performance calculation is performed, from which the design point performance parameters may be determined. The basic characteristic is then scaled according to the resultant design point data. The resultant, scaled characteristic is then transformed into all the alternative representations. A text file is then created in Fortran format containing the data from the scaled and transformed characteristics. This text file is then inserted directly into the gas turbine performance model. This technique ensures that the performance model will produce identical performance predictions when the off-design calculations are applied to the design point.

4.5.3 Out-of-range operation

During a performance simulation, it is possible that the solver will attempt to set conditions which will not produce sensible results. Nevertheless, the performance code should be capable of dealing with these inappropriate conditions. Therefore, it is necessary for guess values of component performance outside the mapped range to produce answers which, while inevitably inaccurate, follow logic such that the iteration may be continued.

The simulation includes the following procedures for handling Betas and non-dimensional speeds outside the mapped range:

- With increasing Beta, the specific torque should become more negative for compressors and more positive for turbines. At both high and low Beta, pressure losses should be high. With decreasing Beta, flow should increase; it should only decrease with increasing Beta until choking occurs.
- Flow and specific torque should increase with increasing speed.
- Flow and pressure loss should not become negative for positive speeds.

4.6 THERMODYNAMICS

4.6.1 RRAP Thermodynamic Routines

Throughout the project, the thermodynamics and compressible flow calculations were performed using the RRAP routines where possible [67].

The RRAP thermodynamics library contains many routines, particularly including procedures for determining ambient temperature and pressure as a function of altitude, calculation of gas constant and specific heat for various gaseous mixtures, enthalpy and entropy calculations, compressible flows and numerous routines for common calculations in gas turbine engines. These routines have been tried and tested over many years.

4.6.2 Rewriting of Routine for Compressible Flow through an Area

The RRAP routine, however, was overridden in the calculation of the compressible flow through an area occasionally, as it sometimes failed to converge on a solution. Therefore, an alternative routine was written to replace that of RRAP when iteration failure was encountered. The details of this revised calculation procedure are shown below.

1. Determine whether flow exceeds choked flow by determining choking flow speed and density, then finding choked flow

2. If unchoked, set three values for guesses of Mach number. These are initially set to 0.2, 0.5 and 0.8.

3. Determine the error resultant from these guesses. This is achieved by calculating the density and static temperature associated with the guessed value of Mach number. The flow speed resultant from the known mass flow rate, the density and the flow area is then calculated. The sonic velocity is found from the calculated value of static temperature, the ratio of specific heats and the gas constant. The flow speed is then divided by the sonic velocity to give a new estimate of Mach number. The error is then found by subtracting the new value from the guess.

4. See if we have errors which bracket the solution. If this is the case, the two guesses which bracket the solution are used and the bisection method is employed to find the correct value of Mach number. If bracketing is not found, the range of the guesses is widened. The process is iterated until a sufficient accuracy is achieved.

As this routine may be used a number of times in an iteration and thus errors may multiply rapidly, it is important that the tolerance used in the iteration is very tight, without being so small that errors within the Mach number calculation itself become significant. As the solution value is always between zero and one and the precision of an eight byte (double precision) real number is around fifteen decimal places [16], a tolerance of 10^{-8} was considered to be a reasonable compromise. As a bisection method is employed, the function is smooth and continuous and, for unchoked flow in the correct direction, a unique solution exists within the limits of Mach number between zero and one, a solution should always be found.

4.7 PROGRAMMING STRUCTURE

4.7.1 Language Choice

As there is a large body of knowledge associated with Fortran within Cranfield University, Rolls-Royce and with the author, the use of this language was attractive. However, the use of Fortran 77 would require the passing of large quantities of variables between procedures or the use of common blocks, both of which are unattractive propositions. Therefore, it was decided to employ the more object-oriented structure of Fortran 90. Therefore, the component characteristics, spool parameters, gas properties and fuel type were stored as user-defined types.

4.7.2 Compiler Choice

Using the features of the chosen compiler, Compaq Visual Fortran v.6.5 (known in previous incarnations as Microsoft Fortran PowerStation and then Digital Visual Fortran), it was intended to be able to use the program as either a stand-alone executable or to be called from the Microsoft Excel spreadsheet software. Therefore, interface routines were built within the Fortran code for the building of a Windows Dynamic Link Library (DLL) and also using Microsoft Excel's Visual Basic for Applications (VBA) programming environment to create user-defined functions which make calls to the DLL. Execution of the code using this approach is slower than using the stand-alone executable. However, large time savings are then achieved in the post-processing of the results, as tables and graphs can be created quickly and easily from Excel.

4.7.3 Linking of Fortran and Visual Basic

4.7.3.1 Introduction

In order to link the Fortran code to the Visual Basic (Applications Edition) code of Microsoft Excel, it was necessary to use syntax which is very poorly documented. In practice, the required syntax and compiler and linker settings were found by an approach of trial and error. For this reason, the required settings are detailed here for the benefit of future researchers. Note that the below method is not the only method which will work. It is, however, tried and tested and thus should produce the desired results without problem.

4.7.3.2 General Considerations

VARIABLES AND TYPES

The most important rule to remember when dealing with mixed language programming, in particular between Fortran and Visual Basic, is that variables must always be the same size. To this end, this researcher always declares floating point variables as 8 bytes and integer variables as 4 bytes, even when this is not necessary for numerical reasons; the impact of this on modern computers is very small and it saves an enormous amount of debugging. It is also important that all variables be properly declared. To this end, the statements "`IMPLICIT NONE`" in Fortran and "`Option Explicit`" in Visual Basic are immensely useful. It should be noted that failure to declare a variable properly within the Visual Basic will usually result in the Fortran code running perfectly, but failing to return any results to the calling routine. This behaviour makes the tracing of bugs rather difficult.

User-defined types can be passed between Fortran and Visual Basic, but care must again be taken to declare the types identically. The passing of strings in user-defined types can cause problems due to the different ways in which they are handled in the two languages and therefore this should be avoided unless absolutely necessary. An example of the Fortran and Visual Basic code for a simple user-defined type is shown below:

```
      TYPE, PUBLIC :: ITCHECK !Iteration check
        REAL*8    VAL1
        REAL*8    VAL2
        REAL*8    ERROR
      END TYPE

Public Type ITCHECK 'Iteration check
    VAL1 As Double
    VAL2 As Double
    ERROR As Double
End Type
```

Arrays are handled slightly differently in Visual Basic and Fortran in respect of the base value of an array. Visual Basic indexes the first array element as zero by default, while Fortran indexes the first element as one. This problem can be circumvented by using the following line in a Visual Basic file or module:

```
Option Base 1
```

However, this author believes that it is better practice to explicitly declare the array bounds for each array. This then permits the copying of a procedure to a different file without the potential problem of the base index changing. This is done with the following Visual Basic code example:

```
DIM SampleArray (1 to 3, 1 to 3) As Double
```

If common blocks or persistent variables have been used within the Fortran, this can create problems. While this author has avoided their use, such a situation is encountered when using some of the RRAP routines. The use of the common blocks requires that the variables in the common block be set to the correct values every time that Visual Basic calls the Fortran code. This is necessary as the order in which Visual Basic calls the Fortran code cannot be assumed, especially where Excel is used.

OUTPUT

When running a DLL, it is not possible to write to a console window. Therefore, the DLL must never run code containing a statement of `WRITE(*,*)`. This can be avoided by either setting a compiler switch to comment out all such lines or by using a variable flag and IF statements to determine whether such code is run.

Furthermore, it is recommended that file writing be avoided when compiling the code into a DLL. File writing greatly increases the computation time and is unnecessary if the results are being returned directly to the calling Visual Basic program or spreadsheet. When file output is required for debugging purposes, this is far easier using the console executable.

4.7.3.3 Fortran Syntax and Settings

The Compaq Visual Fortran environment permits the development of a number of different builds of software within the same project workspace. This is through the use of different projects. This is an important consideration, as it was intended to build both a stand-alone executable, for quick calculation, and a Dynamic Link Library (DLL) for linking into a Microsoft Excel spreadsheet.

To create an environment in which two different builds can be created, first create a blank project workspace within the Visual Fortran Developer Studio. Then create a "Fortran Console Application" project and choose to add it to the current workspace. Similarly, now create a "Fortran Dynamic Link Library" project and also add it to the current workspace. This now gives the two projects.

Once the projects have been created, it is time to add code to them. Working first in the console application project, as it is considerably easier to debug this, start writing the Fortran. The main procedure should be as short as possible and written within a separate file as this will not be required for the DLL. It is important to write code in sub-procedures wherever possible, as it can be useful to only call selected pieces of code from the spreadsheet, for instance to investigate only compressor behaviour without having to run the full simulation.

After the main coding has been completed and the code has been debugged, it can now be altered for use in the DLL. To do this, you first need to add the existing files to the DLL project. To do this, simply drag and drop the source files from one project to the other, holding the CTRL key in order to copy them. Take care not to copy the file containing the main routine.

Now, some settings for the DLL project must be checked and, if necessary, altered. The settings shown in Table 8 are required.

Table 8 - Fortran Compiler Settings for DLL Build

Parameter	Setting
Argument Passing Conventions	Default
External Name Interpretation	Upper Case
Default Real Kind	8
Default Integer Kind	4
Common Element Alignment	1
Structure Element Alignment	Natural

In order to export a procedure, a compiler directive must be used. This directive takes the form demonstrated by the example below:

```
      !DEC$ATTRIBUTES DLLEXPORT,ALIAS:'DLLFULLSIMUL'::DLLFULLSIMUL
```

Here, the `!DEC$ATTRIBUTES` indicates that the line is a compiler directive. Then, the DLLEXPORT shows that the routine is to be exported. The statement `ALIAS` is not essential, but this author believes it to be good practice. The name within the inverted commas is the name by which the routine will be exported. Finally, the name after the double colons is the name of the routine within the Fortran code. As the `ALIAS` statement also changes the name of the routine when being called from within the Fortran, it is necessary to create a shell routine. Thus, in order to call the routine "`FULLSIMUL`," the shell routine "`DLLFULLSIMUL`" is created:

```
      SUBROUTINE DLLFULLSIMUL(ALT,FLTSPD,WF,RH,SWSTART,SWHYDRA,
     + CHICTYPE)
      !DEC$ATTRIBUTES DLLEXPORT,ALIAS:'DLLFULLSIMUL'::DLLFULLSIMUL
C Inputs
      REAL*8    ALT                         !Altitude
      REAL*8    FLTSPD                      !Flight speed
      REAL*8    WF                          !Combustor Fuel Mass Flow
      REAL*8    RH                          !Relative humidity
      INTEGER*4 SWSTART
      INTEGER*4 SWHYDRA
      INTEGER*4 CHICTYPE
      CALL FULLSIMUL(ALT,FLTSPD,WF,RH,SWSTART,SWHYDRA,CHICTYPE)
      END SUBROUTINE
```

4.7.3.4 Visual Basic Syntax

Within the Visual Basic code, the DLL routines to be called must be declared as in the following example:

```
Declare Sub DLLNOZZLEQ Lib "path and filename of DLL" (PAMB As Double,
PTOT As Double, TTOT As Double, W As Double, Q As Double)
```

If the DLL is placed within the Windows path (e.g. C:\WINDOWS\ or C:\WINDOWS\SYSTEM\), it is not necessary to specify the path of the DLL. However, it is good practice to do so, as different versions of a file may exist on the system and one must always be sure of using the correct copy.

Once the routine and any user-defined types have been declared, the routine is called as though it were a native Visual Basic routine. However, great care must be taken to ensure that all variables are properly declared.

When calling a routine with an array as one of the parameters, the Fortran routine is declared as normal. However, in the Visual Basic code, the parameter is declared as a scalar. The calling code then passes the first element of the array.

4.8 RESULTS

The simulation was tested on a large number of windmilling points, over a range of flight speeds and altitudes. The model achieved convergence in most cases, though failures were occasionally detected. These convergence failures were rarely encountered when the proposed characteristic representations were used. With the other methods of representation, the convergence failures are believed to be due to the simulation producing changes in flow conditions across the components which were larger than those which were produced with the proposed methods of component representation when extreme guesses of betas were made. This then led to a bumpiness in the errors and thus presented a problem for the Newton-Raphson solver. It should be noted here that alternative types of solver may behave more robustly in such conditions.

When the proposed methods of turbomachinery characteristic representation were used, the simulation produced realistic solutions for all the windmilling points tried. It is therefore considered that the solver strategy was successful.

4.9 SUMMARY

In order to study the effect of changing the component characteristic representations, a gas turbine performance model was built. RRAP routines were used where possible, in order that the model could be incorporated into Rolls-Royce's systems with a minimum of problems. The sub-atmospheric nozzle case was considered and a model developed, as this is encountered regularly when performing the flow matching iteration for windmilling conditions. A solver scheme was created, using a nested iteration. The resultant model was tested for robustness at a number of windmilling points and produced converged solutions for steady-state windmilling in nearly all cases. Therefore, the design of the performance model was considered to be successful.

5 Modelling of Other Components and Phenomena

5.1 INTRODUCTION

Previous sections of this thesis have discussed the modelling of the compressors, turbines and nozzles of the gas turbine engine for the study of windmilling and relight. This section investigates the performance of the other components of the engine. The components examined here are the combustor, the power offtake systems, the starter, the bearings, the intake, the mixer, internal flows and the nozzle loss.

5.2 COMBUSTION

5.2.1 Introduction

Although the main focus of this thesis is on modelling the turbomachinery and solver scheme, some studies were carried out related to the ignition and pull-away behaviour of the combustion system. The majority of these studies have been carried out by M.Sc. students, managed by the author of this thesis.

5.2.2 Combustor Loss

During the above-idle operation of the gas turbine engine, the combustor experiences a loss in total pressure from two different mechanisms. The first of these is termed the "cold loss" and is simply the aerodynamic drag effect of flow over a surface and around an obstruction. The second is termed the "hot loss" and is due to the expansion of the working fluid in the combustion process.

When the engine spools down and windmills, the hot loss drops to zero as the combustion process stops. There is still a hot loss until a steady state is reached due to the heat soakage from the combustor to the airstream, although this is so small that it may be neglected.

During the windmill process, the cold loss is, however, the same as during above-idle operation. This pressure loss is often regarded as being proportional to the total pressure of the combustor inlet air [46]. However, as this is an aerodynamic loss, it would be more appropriate to regard the cold loss as being proportional to the dynamic pressure of the air at a reference area [44].

5.2.3 Ignition Loop

In Figure 11, a typical ignition loop was presented. This is determined through a combination of correlations through previous experience and through testing. However, a number of problems may be encountered in trying to use the ignition loop directly in a relight model. These issues are explored in Sections 5.2.4 and 5.2.5.

5.2.4 Effect of Fuel Temperature on Ignition Performance

5.2.4.1 Introduction

During a visit by the author of this thesis to the DERA Pyestock Altitude Test Facility (ATF) to observe relight testing, it was noted that, during the course of a day's testing, the fuel and oil temperatures increased continuously, from start values at those of the air on the ground. Thus, for those tests conducted towards the end of the day, the temperature of the fuel supplied to the combustor was very high. Indeed, the fuel temperature was observed to rise to 345K and the oil temperature to 350K. This is in contrast to the temperature of the fuel supply to a windmilling engine. At altitude, the fuel tanks are at a lower temperature than on the ground. Then, if the engine windmills for a significant time, the oil temperature drops due to the use of an oil cooler. Then, the fuel which passes through the preheating heat exchanger in the oil system will not be preheated as much. Hence, the fuel supplied to the combustor will be substantially cooler than in the ATF. Meanwhile, some combustion tests are performed with fuel supplied directly from fuel tanks which are at ambient temperature. Therefore, it was decided to study the effect of these differing fuel temperatures on the ignition capability of the combustor.

5.2.4.2 Fuel Temperature Effect on Ignition [25]

The following work was performed by Cyrus Haghrooyan, an M.Sc. student at Cranfield University, supervised by Dr. Pilidis and managed by the author of this thesis.

Haghrooyan investigated the effect which a change in fuel temperature had on the ignition loop of a combustor. His analysis started by examining the range of fuel temperatures encountered in testing and in-service relighting of an engine. Then he examined the effect which a change in the fuel temperature had on the surface tension, density and viscosity of the fuel and investigated the changes in droplet sizes produced by fuel injection. The resultant changes in ignition capability were then estimated using a minimum ignition energy correlation produced by Lefebvre (Eq. 2).

Using this method of analysis, Haghrooyan produced modified ignition loops for hot fuel (ATF test) and for cold fuel (cold-soaked windmilling engine). One such modified ignition loop is shown in Figure 65.

5.2.4.3 Analysis and Implications

ANALYSIS

The work clearly shows that the temperature of the fuel injected into the engine will greatly affect the ability of the engine to relight. As tests in an ATF are conducted with high fuel temperatures, it is likely that this will produce optimistic results for the relight envelope. Conversely, for an aircraft such as the Nimrod radio surveillance aeroplane, which cruises with only two of its four engines powered, the actual relight envelope may be expected to be substantially worse as the unpowered engines will be very cold.

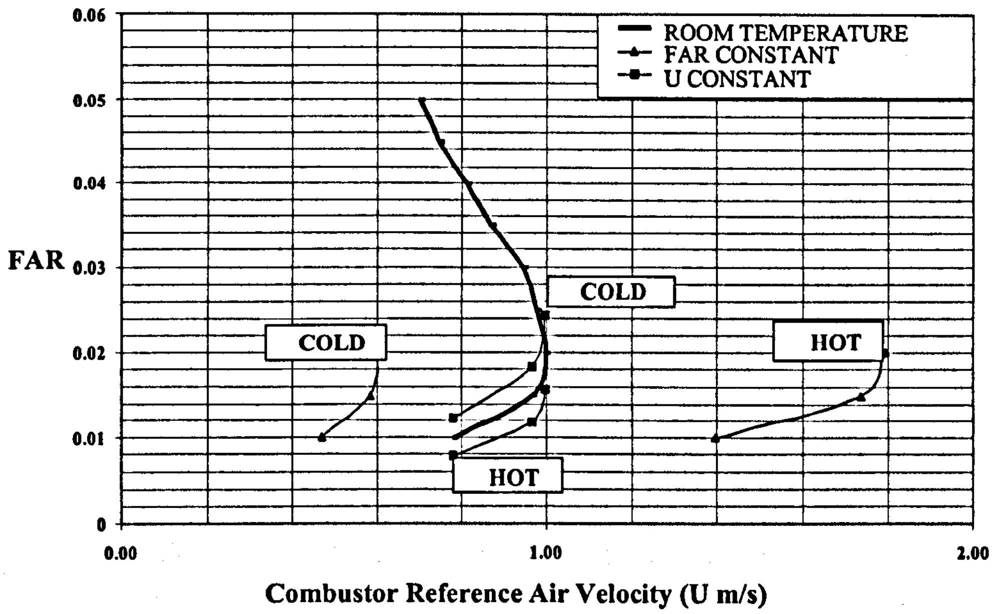

Figure 65 - Combustor Ignition Loop Variation with Fuel Temperature [25]

However, the results of this work should be treated with a little caution, as they have been produced from only one set of correlations, without the experiment that should now be conducted. Nevertheless, the relationship which produces larger droplet sizes and lower temperatures is reliable, as is the relationship about the adverse effect which this will have on ignition performance.

It should also be noted that the same work conducted by Haghrooyan can be applied to analysis of combustion efficiency at pull-away conditions and that similar adverse effects may be expected with a low temperature fuel supply.

Implications for Engine and Aircraft Design

As the temperature of the oil in a flying windmilling engine may be expected to drop reasonably quickly, and hence the fuel temperature would drop quickly also, it may be worth considering cross-bleeding fuel from a hot engine to a windmilling one to aid relight (Figure 66).

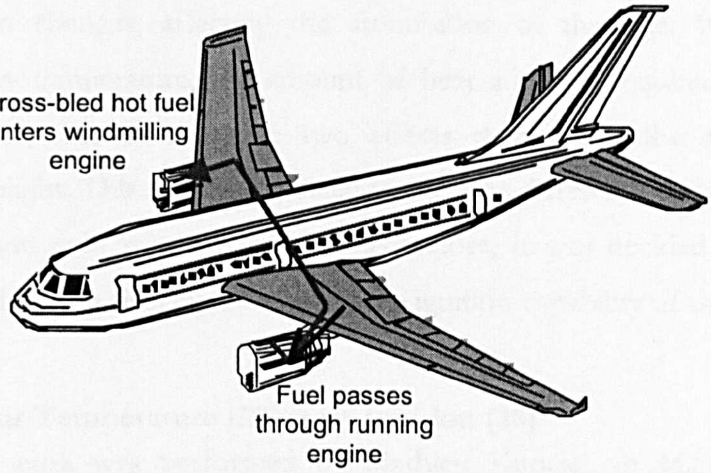

Figure 66 - Fuel Cross-Bleed from Hot Engine to Aid Relight

Alternatively, the oil cooler should be bypassed when the engine is windmilling in order to retain the oil's heat.

IMPLICATIONS FOR ENGINE TESTING

If gas turbine performance modelling can achieve high levels of reliability, the purpose of testing on a test bed can be largely reduced to verifying that the performance modelling produces good results, performing some calibration of the model and doing any troubleshooting necessary. In order to achieve this, it must be possible to run the performance synthesis with identical conditions to the tests conducted on the test beds. One of the factors which have been shown to be important for relight modelling is the temperature of the fuel supply. Thus, any performance model used for calibration against the test bed should include a model of how the ignition loop changes with fuel temperature.

5.2.5 Effect of Air Temperature on Ignition Performance

5.2.5.1 Introduction

During the spool-down of an engine after a flame-out, the air temperature entering the combustor drops, quickly at first due to the power decrease and deceleration of the compressor system, then more slowly as a result of heat soakage from the compressors. As this then affects the density of the air entering the combustor, the flow speed also changes, affecting the atomisation of the fuel. With reducing combustor entry temperature, the amount of heat addition required for a flame kernel to develop increases. These two effects then affect the ability of the combustor to relight. This may be expected to produce different behaviour between quick relights and cold windmill relights. Therefore, it was decided to study the effect of these differing air temperatures on the ignition capability of the combustor.

5.2.5.2 Air Temperature Effect on Ignition [38]

The following work was performed by Mathieu Kupcic, an M.Sc. student at Cranfield University, supervised by Dr. Pilidis and managed by the author of this thesis.

Kupcic attempted to investigate the effect which a change in air temperature would have on the ignition performance of the combustor. He started by performing a study of the heat soakage within the compressor system during spool-down with a view to predicting the change of combustor inlet temperature over time. The effect which this change in air temperature had on the atomisation process was then studied, using correlations for the fuel injector performance. He also built upon Haghrooyan's work to investigate the change of fuel temperature throughout a flight. The air and fuel temperatures resultant from these analyses are then incorporated into the minimum ignition energy correlation which Haghrooyan used.

During the project, Kupcic found that, with a decrease in air temperature, the droplet size decreased. This then has a tendency to make ignition easier. Counteracting this, more energy is required to raise the temperature of the air, although this effect is not described in the correlation used by Lefebvre. The viscosity of the air also increases, which has a negative effect on ignition performance. The net result of the competing effects was, in this work, shown to produce better ignition performance with colder air. This result is counter- intuitive, although it is possible that this could be the case. Further work is required here to check this finding.

5.2.5.3 Analysis and Implications

This study showed that the atomisation process is likely to be improved by lower temperature air, as the shear on the fuel film on the injector will increase with the lower density of the air. However, this effect is dependent on the behaviour of the injector.

Conversely, lower air temperatures mean that more energy has to be supplied to the flame kernel for it to develop.

In this study, it was found that the effect of improved atomisation with a drop in air temperature was dominant. However, this could change with a different combustor design.

As with the work of Haghrooyan, the results of this study should be treated with a little caution, as they rely on the use of only one set of correlations and have not been compared with experiment.

5.2.6 Combustor Heat Transfer

5.2.6.1 Introduction

Although a pull-away issue and thus outside the main focus of this work, it was decided to look at the subject in order to provide some foundation for future continuation of this work. One of the aspects of pull-away performance which is of interest it the possibility of the flame extinguishing while the kernel is still developing. Another area for investigation is the heat transfer to the turbine blades during the initial acceleration of the engine, as overtemperature conditions can easily occur here. Both of these effects are influenced by the heat transfer within the combustor and therefore this effect was studied.

5.2.6.2 Combustor Heat Transfer at Ignition [5]

The following work was conducted by Chris Allan, an M.Sc. student at Cranfield University, supervised by Prof. Pilidis and managed by the author of this thesis.

Allan studied the effects of heat transfer on the combustor during the ground starting of an engine.

In order to provide the inputs to his work, he used the principle of gas turbine performance simulation to predict the temperature, pressure and mass flow rate of the air entering the combustor as a function of time. Compressor and turbine maps were scaled and extrapolated and a transient analysis was conducted for the starting procedure. The combustor geometry and size were determined through a combination of measurement and calculation. The primary zone air temperature is then calculated and the radiative and convective heat transfer is examined.

The study showed that the heat transfer taking place in the combustor can drop the TET by as much as 10 K during the start-up acceleration. However, the rate of heat transfer has been shown to be dependent on a number of factors. Particularly, the composition in the primary zone is important. This is difficult to determine as soot formation is traditionally a problematic process to model and is important here.

5.2.6.3 Analysis and Implications

It was shown that the combustor heat transfer during the acceleration can be important, significantly affecting the turbine inlet temperature. The study was not extended to the effect on the flame kernel development, though the effect is likely to be important here also. Therefore, further studies should be pursued.

5.2.7 Combustion Efficiency

The study of combustion efficiency during pull-away was considered outside the main scope of this project. However, the work described in Sections 5.2.2 to 5.2.6 provides a good starting point for such studies.

5.2.8 Blow-Off

In the course of this study, blow-off of the flame after ignition has not been investigated.

5.3 ACCESSORY DRIVES AND MECHANICAL LOSSES

5.3.1 Introduction

One of the effects which is of great influence in the steady-state and transient windmilling of an engine, especially at low flight speed and high altitude, is the influence of the gearbox drive systems. Particularly, it has been noted that the influence of the offtake systems on the high pressure shaft of the engine can produce a multiplicity of solutions to the matching constraints. Therefore, it was decided to study this area in more detail.

5.3.2 Modelling of Accessories [78]

The following work was performed by Omar Vielma, an M.Sc. student at Cranfield University, supervised by Prof. Pilidis and managed by the author of this thesis.

Vielma worked to investigate the behaviour of the components of the offtake systems of the engine, with a view to incorporating this into the performance model described in Chapter 4.

5.3.2.1 Gearbox

The design point gearbox efficiency was found to be high. An off-design gearbox loss relationship was then found.

5.3.2.2 Hydraulic Loads

Following an investigation of the operation of the hydraulic and oil pumps, it was possible to derive torque - speed representations as shown in Figure 67 and Figure 68. It is noted that hydraulic loads are the largest demand on the offtakes.

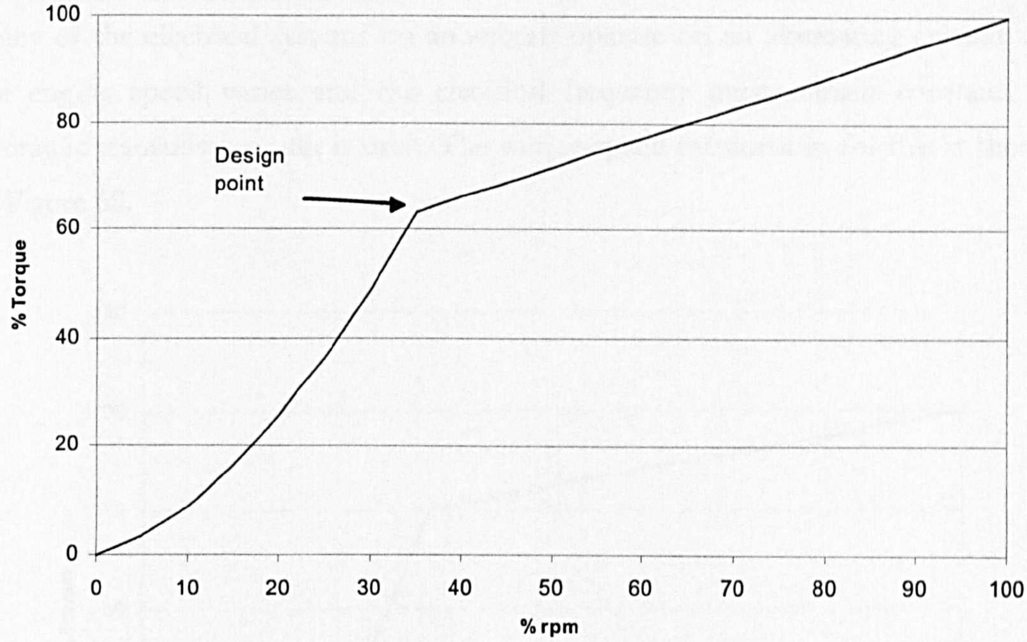

Figure 67 - Torque-Speed Relationship for an Oil Pump [78]

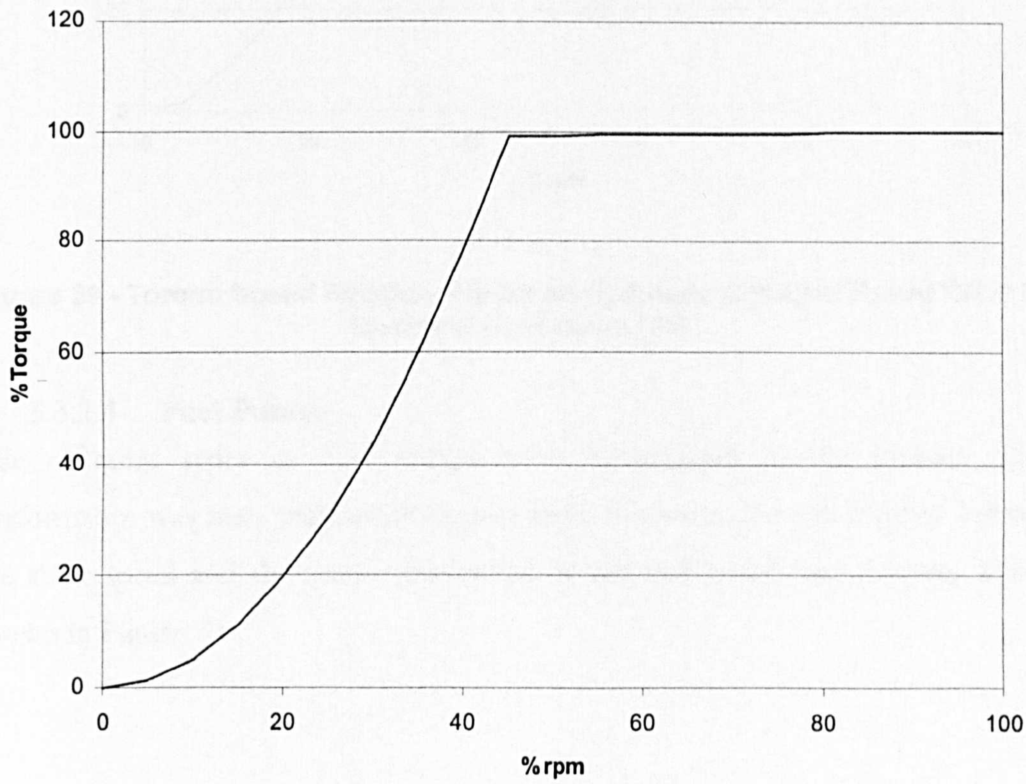

Figure 68 - Torque-Speed Relationship for an Hydraulic Pump [78]

5.3.2.3 Electrical Loads

Many of the electrical systems on an aircraft operate on an alternating current. As the engine speed varies and the electrical frequency must remain constant, an hydraulic transmission unit is used. The torque-speed relationship for this is shown in Figure 69.

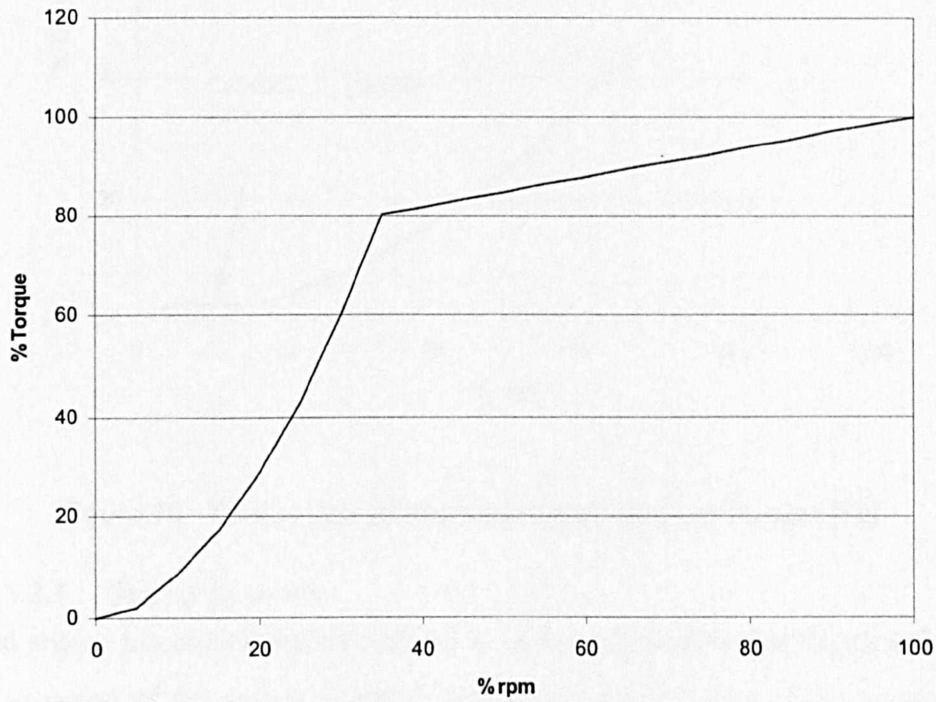

Figure 69 - Torque Speed Relationship for an Hydraulic Constant Speed Drive for Electrical Generation [78]

5.3.2.4 Fuel Pumps

The different types of fuel pumps were investigated in the project. Their performance was then analysed. Resultant from this work, the relationship between the shaft speed and the torque demanded by the fuel pump was derived. This is shown in Figure 70

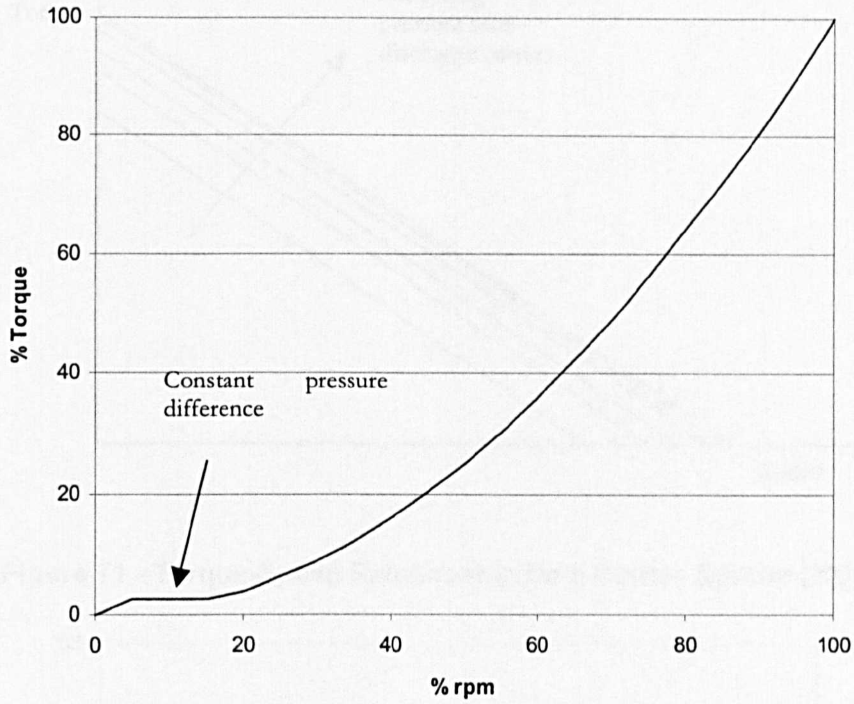

Figure 70 - Torque-Speed Representation for Fuel Pumps [78]

5.3.2.5 Starter Systems

Typical engine pneumatic starter systems were investigated. It was decided that the initial engaging of the starter system is not significant in terms of the acceleration time of the engine, although it is a consideration if the stiction of the engine is high. The relationship between torque and speed was shown to be linear and to vary with starter pressure ratio as shown in Figure 71.

5.3.2.6 Bearing Losses

The majority of the bearing losses were found to be between the rolling elements and the oil. An empirical relationship for the torque was found and the resultant relationship is shown in Figure 72.

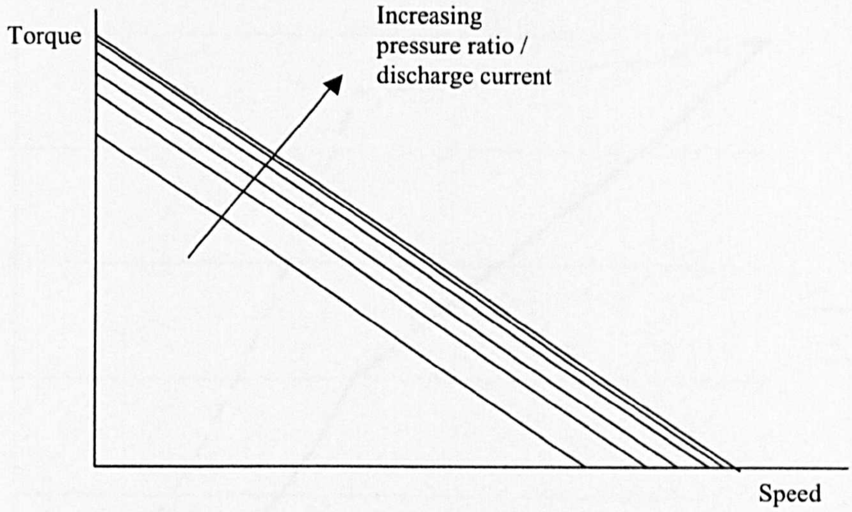

Figure 71 - Torque-Speed Relationship for a Starter System [78]

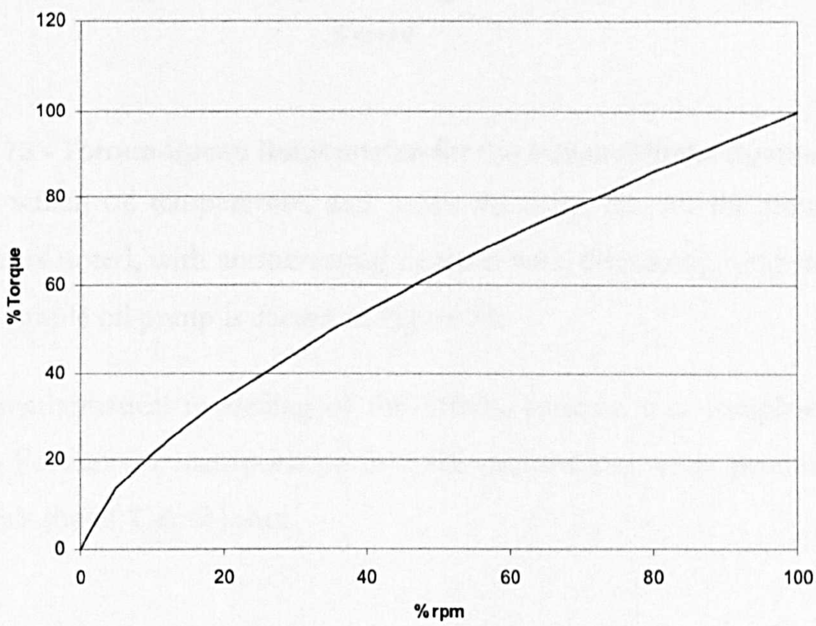

Figure 72 - Torque-Speed Relationship for Bearing Losses [78]

5.3.2.7 Overall Effect

An overall torque-speed relationship for the offtake systems can be created through the summation of the individual relationships. This is shown in Figure 73.

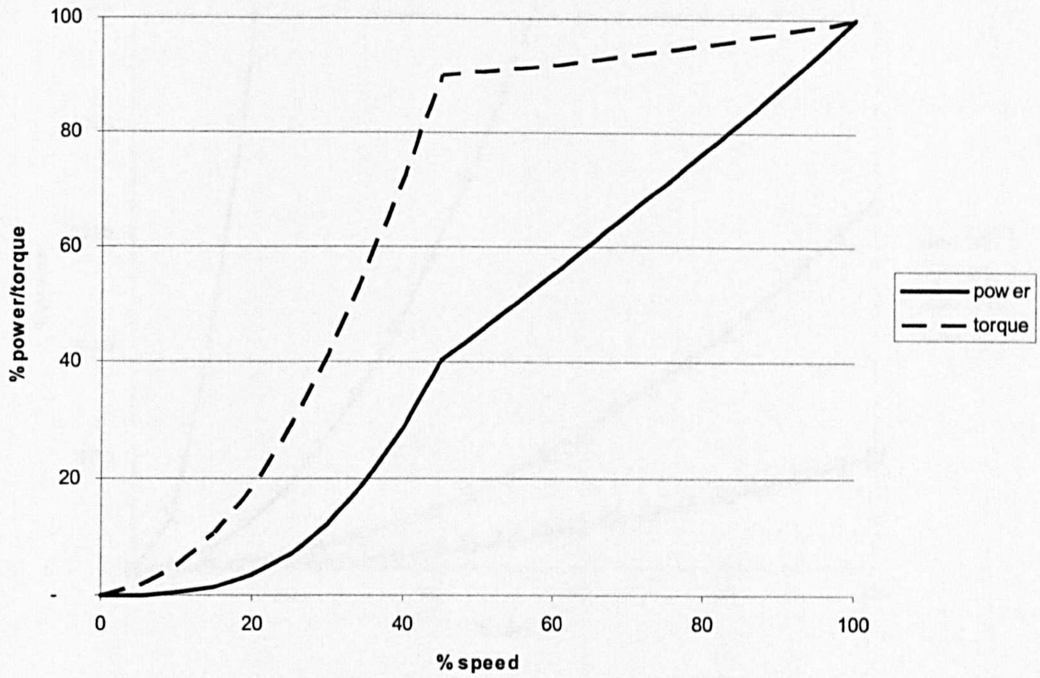

Figure 73 - Torque-Speed Relationship for the Power Offtake Systems [78]

The effect which oil temperature, and hence viscosity, has on the power offtake requirement is noted, with an increasing demand with decreasing temperature. The effect on a simple oil pump is shown in Figure 74.

Once the mathematical modelling of the offtake systems was complete, this was coded into Fortran for incorporation into the performance code produced by the author of this thesis, Geoff Jones.

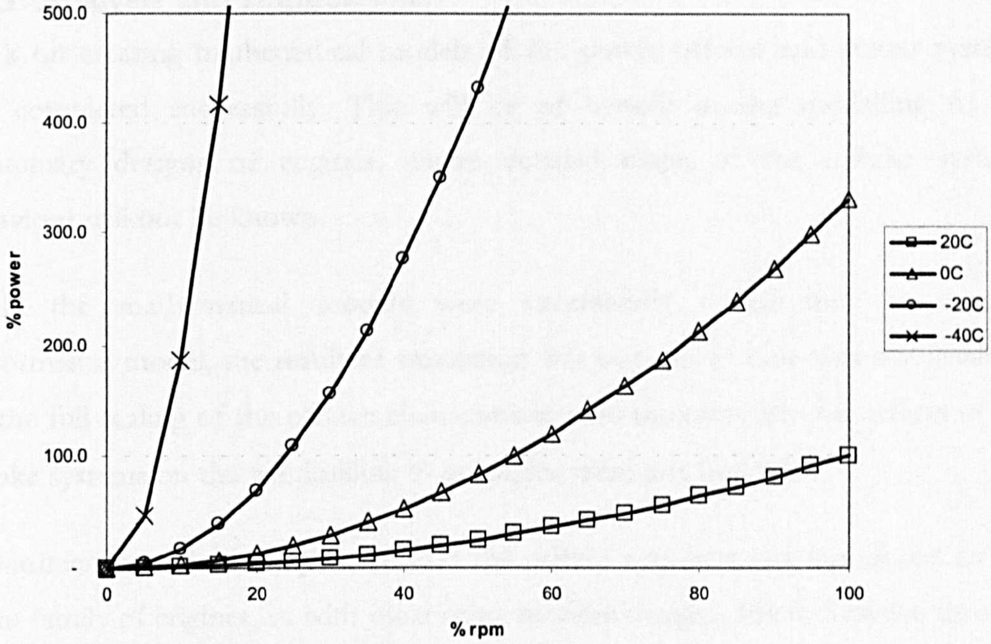

Figure 74 - Effect of Oil Temperature on the Power Requirement of an Oil Pump [78]

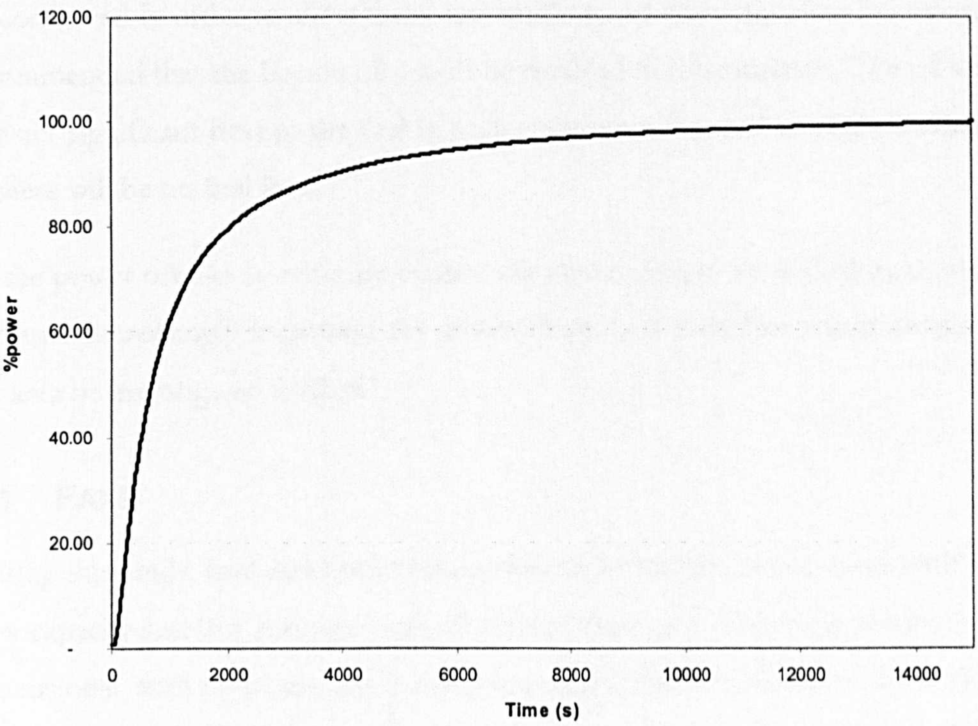

Figure 75 - Variation of Power Offtake Requirement as a Function of Time Following a Flame-Out [78]

5.3.3 Analysis and Implications

Work on creating mathematical models of the power offtake and starter systems was completed successfully. This will be of benefit in the modelling of the preliminary designs of engines, where detailed maps of the offtake systems behaviour will not be known.

While the mathematical models were successfully coded into the overall performance model, the resultant simulation was not run as time was not available for the full scaling of the offtake characteristics and therefore the full effects of the offtake systems on the windmilling of an engine were not studied.

The influence of fluid temperatures on the offtake requirements was noted. In the Trent family of engines, as with most other modern designs, the fuel passes through the oil system before entering the engine. In this way, the fuel absorbs heat, aiding atomisation. The oil system rejects some of its heat to this flow and some to the bypass air. Subsequent to flame-out, it is important that the heat of the oil systems be conserved in order to allow faster windmilling and easier relights. Therefore, it is recommended that the bypass oil cooler be disabled in this situation. The oil will not transfer significant heat to the fuel in such circumstances until a relight is attempted as there will be no fuel flow.

As the power offtake systems are continually demanding more and more power, this becomes increasingly important for windmilling. It is therefore recommended that the area be investigated further.

5.4 FANS

During this study, fans have been considered to be merely compressors with a large flow capacity and low pressure ratio. However, their performance is somewhat two-dimensional, with radial variations being important. This will continue to be the case for windmilling, and indeed the effect is likely to be more pronounced with the large bypass ratios encountered. Work is continuing within the UTC on fan two-dimensional modelling. A number of papers have been published on two-dimensional fan modelling by members of the UTC [86],[87] and [88].

5.5 INTAKES

During the course of this project, intakes have been regarded as lossless. This is a simplification which is considered unimportant in determining the reliability of the solver system, but is important in the modelling of a real engine. Walsh and Fletcher [82] provide some information on the modelling of intakes for gas turbines. The performance at the low flow rates encountered in windmilling is assumed to be the same as for above-idle, though this should be verified.

No work transfer occurs in the nozzle and temperature changes are very small, making heat transfer effects negligible.

5.6 NOZZLES

In terms of the nozzle operation, the nozzle has been modelled in Section 4.4.5. However, the losses within the nozzle were not considered. Pressure losses are generally small [82], though as the pressure driving the engine during windmilling is small, these should be considered.

No work transfer occurs in the nozzle. However, the temperature change when dropping from powered flight to windmilling conditions will lead to heat transfer from the nozzle to the flow. Nevertheless, the nozzle metal thermal capacity is relatively low and therefore this effect can probably be considered to be small.

5.7 MIXERS

As mixers feature in relatively few of today's Rolls-Royce gas turbines, these have not been considered in this study. With the large change in bypass ratio encountered when windmilling, the mixers will not perform as they do at normal running conditions. Therefore, modelling of their behaviour is needed for incorporation into a windmilling model.

5.8 HEAT TRANSFER

The effect of radiative heat transfer was studied by Chris Allan as part of this work (see Section 5.2.6.2). Also, some compressor heat transfer analysis was performed by Mathieu Kupcic to provide input to his analysis (see Section 5.2.5.2).

However, heat transfer within the compressor and turbine systems has not been studied in detail, although it is recognised that this will greatly influence the behaviour of the engine for quick relight and for pull-away. As the focus of this project has been on steady-state behaviour, this does not affect this thesis. Work is taking place within the Cranfield UTC [57] studying heat transfer at above-idle conditions and it is anticipated that this could be extended to the relight problem.

5.9 VOLUME PACKING

Volume packing is the effect of an acceleration increasing the density of the working fluid and thus producing an imbalance between the flow into and out of a component. While this is sometimes considered to be an important transient effect, it may be considered of relatively minor influence in the performance of the gas turbine at windmilling and relight conditions.

5.10 BLEED FLOWS

Bleed flows should be modelled as described in Walsh & Fletcher [82] for above-idle performance analysis. This does not change for sub-idle work.

5.11 INTERNAL FLOWS

While the flows of main interest in gas turbine performance modelling are those of the two main airstreams of the core flow and the bypass flow, the internal flow of the engine can also be of importance. This is usually relatively small, but can increase under certain conditions.

One important use for the internal flow is in providing cooling air for the turbines. During relight, overtemperature conditions sometimes occur. Therefore, the reliable supply of cooling air to the turbines is important.

The internal flow also controls the heat transfer to and from the disks. This flow is complex and is being studied within the UTC by other researchers [57].

The rate of internal flow has in the past been controlled largely by seals. During transient operations, these seals dilate and contract due to the movement of the rotor through heat soakage and centrifugal effects. However, more recently, these flows have been controlled by choked holes. These are more reliable as they do not exhibit this behaviour.

5.12 SUMMARY

Modelling has been carried out on the combustion and power offtake systems. The other systems of the gas turbine engine have been considered briefly.

The ignition behaviour of the combustor was considered and studies were carried out investigating how this is affected by changes in the fuel temperature and the air temperature. The heat transfer in the combustor at start-up was also considered.

The relationship between the engine speed and the torque requirement of the power offtake and starter systems was investigated in some detail. Methods of representing the systems were created and the effect which fluid temperatures had on the performance of those systems was examined.

6 Benefits of the Research

6.1 INTRODUCTION

Rolls-Royce has identified altitude relight and windmilling as an area in which its understanding and analytical capability is less than required. This section investigates the effect which this work will have upon the company by investigating the current design and testing process and the reduced development costs and risk which improved windmilling and relight modelling bring.

This section explores the different stages of design and development within Rolls-Royce and the roles for modelling and testing. The information in this section has been gathered through talks with staff at Rolls-Royce and a lecture by Prof. Howse [28], also of Rolls-Royce.

6.2 PRODUCT DEFINITION LIFECYCLE

The product development process at Rolls-Royce follows a four stage model, outlined in Figure 76.

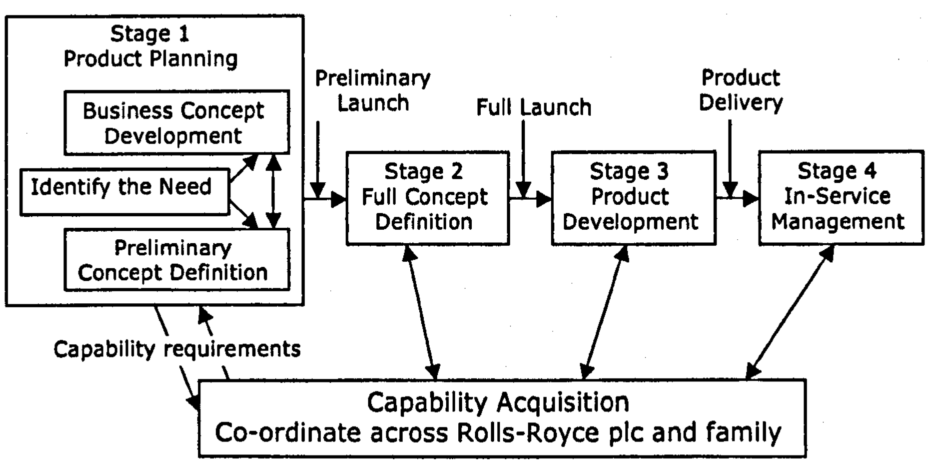

Figure 76 - Product Definition Lifecycle at Rolls-Royce [28]

Stage 1, Product Planning, covers the basic conceptual design for an engine and the business case for its development. This stage involves simple performance modelling, largely covering the design point thrust requirement, fuel consumption and emissions and basic thermodynamic cycle. Some preliminary component models may be available to allow off-design performance modelling.

Stage 2, Full Concept Definition, covers much of the preliminary design. At this stage, predictions of the performance of the components will be made, although full detail designs will not be available. These preliminary predictions are termed "component bids." These bids may then be used for performance analysis of the off-design behaviour of the engine. This will then allow many potential problems with the proposed design to be identified before the component designers commit large quantities of resources to a fundamentally poor design.

During the process of Stage 2, better models of component behaviour will become available, through analysis of the component designs. This then allows performance modelling accuracy to improve. By the end of Stage 2, the following should have been modelled by the performance engineers:

- Cooling air flows
- Sealing air systems
- Bearing loads
- Bleeds
- Overspeed condition
- Shaft breakage response
- Overheat
- Windmilling and relight analysis

Before the project can move to Stage 3, a detailed plan for the design and certification programme must be available.

Stage 3, Product Development, includes the specification of the testing programme, the testing itself and the analysis of those tests. After both component tests and early engine tests, the characteristic performances of the various components may be tweaked, improving the accuracy of the performance model. After the development testing follow flight tests and certification.

At Stage 4, In-Service Management, the engines enter service. The engines are then monitored in respect of fleet performance, performance deterioration and service incidents.

6.3 CURRENT DESIGN AND DEVELOPMENT PROCESS WITHIN ROLLS-ROYCE

6.3.1 Introduction

Windmilling and relight modelling are used at many stages of the development process. Even at the concept stage of a revolutionary design, airframe manufacturers are interested in the windmilling drag which an engine produces. Then, analyses of relight and other windmill performance synthesis are conducted throughout the design process.

6.3.2 Preliminary Design

During Stage 1 of the design process, a preliminary design of the engine is produced. This will include such design parameters as component sizing and engine layout. Values will be set for the thrust requirement, acceptable gaseous emissions level, fuel consumption and possibly noise limits. In order to be able to specify the basic design parameters, a study of the engine's thermodynamic cycle will be required. Using estimates of component efficiencies from previous experience of similar engines, a design-point study of the engine performance can be developed. Using estimates of off-design performances of the components, it is also possible to examine some off-design behaviour of the whole engine. This can be of use in studying the emissions over the taxi, take-off and landing cycles, where the engines operate at different power settings.

6.3.3 Engine Modelling

6.3.3.1 Introduction

During the design and development process, numerical and computational modelling performs an important role, reducing design risk without the commitment of large resources into expensive testing. This section briefly describes some of the modelling methods used.

6.3.3.2 Component Modelling

This section briefly describes some of the modelling methods used for the development of the main thermofluid components of a gas turbine.

During the design of compressors, combustors and turbines, Computational Fluid Dynamics (CFD) is used extensively. In the turbomachinery components, this is used to predict design point performance of the whole component and off-design performance of parts of the components, such as turbine blade cooling. Full analysis of the off-design performance of a whole compressor, particularly at off-design values of incidence, is currently not practical, both due to the computational time required and the accuracy of modelling the separation process [76]. CFD analysis in combustors is largely focused on emissions prediction.

6.3.3.3 Engine Performance Modelling

This section gives a brief description of engine performance modelling. Further discussion of the role of performance modelling in engine design and development can be found in Sections 1.2.3, 1.3.4.2, 2.3 and 2.4.

During Stage 2 of the project, the off-design performance of the engine is investigated in considerably more detail than in Stage 1. This process is fed by the availability of improved models of the component performances and delivers specifications for the development of other components, while examining the overall performance of the engine.

Through modelling the engine behaviour at the extremes of full power, windmilling and ground starting, the extreme loads acting on the engine may be studied. This then provides information to the designers of the mechanical systems, such as bearing designers, and to the airframe manufacturer. By investigating the nozzle flow conditions and the fan operating point, estimates may be made of the engine noise throughout the flight envelope. Emissions may be estimated through the study of the combustor flow conditions. To specify a starting system for the engine, predictions of the starting torque requirements are needed. The control engineers require good models of the off-design behaviour of the engine in order to specify variable geometry and fuel schedules. Finally, it is possible to investigate the engine operation at some extreme conditions of operation, which, while not commonly encountered, must be considered and some of which are certification issues. These include the operation of the engine subsequent to a bird strike and the windmilling and relight capability of the engine.

In order to investigate such issues, it is important to use good models of component behaviour.

6.3.4 Engine Testing

6.3.4.1 Introduction

During the development of an engine, computational methods are used where possible to predict the performance of components. Due to Rolls-Royce's design philosophy for the Trent family of engines of making incremental changes between models, these techniques may be expected to work reasonably well. However, tests must always be conducted to ensure that the components, and the engine, behave as expected. Such tests may frequently produce behaviour slightly different from the predictions, justifying their use. From these tests, it is possible to improve the computational models, with the tests effectively calibrating them. The tests may also show previously unconsidered problems. Testing takes place primarily during Stage 3 of a project.

This section investigates briefly some of the testing methods used in the development of an engine.

6.3.4.2 Component Testing

INTRODUCTION

This section describes some of the main tests conducted on the major aerothermodynamic components of the engine: the compressors, turbines and combustors.

COMPRESSOR TESTING

Once a compressor has been designed and a prototype built, it may be tested. Compressor testing is usually conducted on a rig, whose basic layout is as shown in Figure 77. On this rig, the compressor is operated at a range of rotational speeds and throttle openings. This effectively varies the compressor operating point in terms of non-dimensional speed and non-dimensional exit flow. Measurements are made at the front and rear of the compressor of the temperature, pressure and Mach number. These values can be found both as bulk averages and also as radial and circumferential profiles. Measurements may also be made in between the blade rows. The metal temperatures and stress levels are usually also monitored. Through the generation of a number of test points, a compressor characteristic may be built.

Because of considerations of instrumentation accuracy, the operating range of the motor and the throttle and the hourly cost of running such a rig, the compressor performance is normally only mapped at speeds above around 60% of design speed.

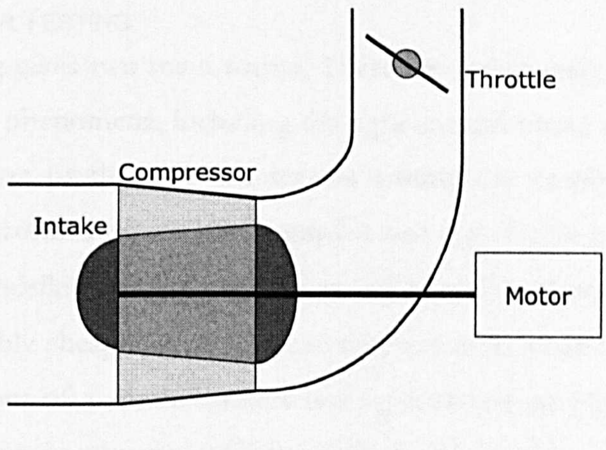

Figure 77 - Compressor Test Rig

TURBINE TESTING

As with the compressor, a turbine may be tested at a range of speeds and pressure ratios. A typical rig design is shown in Figure 78. Unlike a compressor test, a turbine is rotated by the airstream and its speed is controlled by a brake. The pressure ratio across the turbine is determined by extraction compressors which produce sub-atmospheric conditions at the turbine exit. Similar measurements are made as for the compressor.

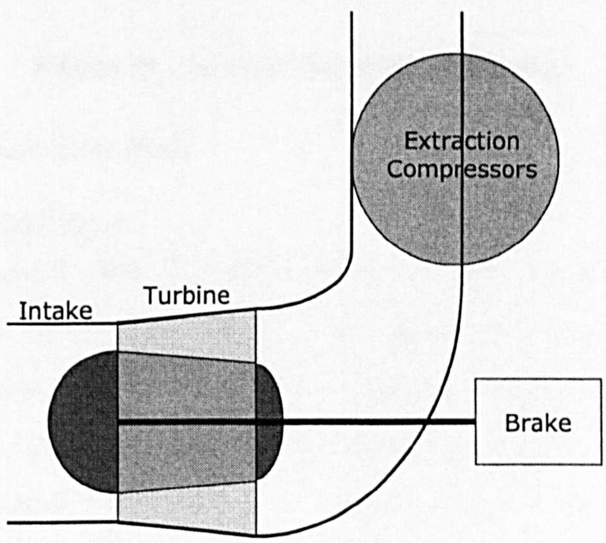

Figure 78 - Turbine Test Rig

Combustor Testing

Combustor testing takes two main forms. These are sector tests and annulus tests. Many combustion phenomena, including the light-around phase of starting, depend upon the interaction of the different sectors around the combustor annulus and must therefore be conducted on whole-annulus test rigs. Other phenomena may be reasonably well modelled using a single sector or a small number of sectors. Sector tests are considerably cheaper to conduct as they use a far smaller supply of air and fuel. A typical layout of a whole-annulus test rig is shown in Figure 79. Such a rig may be used to study combustion efficiencies and emissions production. While it is possible to perform ignition and some stability testing on such a rig, it is difficult to perform full starting tests as the rig control systems will not react in the same way as an actual engine.

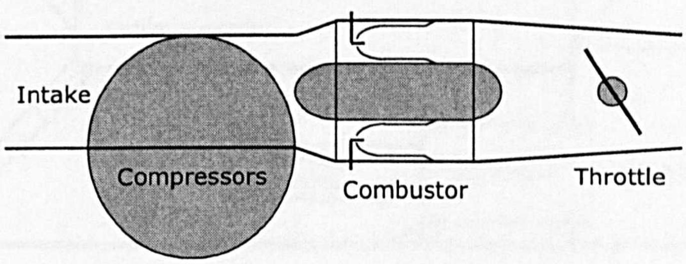

Figure 79 - Annular Combustor Test Rig

6.3.4.3 Engine Test Beds

Outdoor Test Beds

Outdoor test beds are the simplest test beds for an engine. These are of the form of a simple cradle to support the engine. The engine may then be instrumented according to the test requirements. While a range of tests could potentially be conducted on such a rig, the requirements for wind-free, dry weather and local noise considerations effectively prohibit their use for many tests. However, water ingestion and bird strike tests are usually conducted outdoors.

Indoor Sea Level Test Beds

Indoor test beds are the most commonly used engine test beds. A typical layout for a sea level test bed is shown in Figure 80. These are closely controlled environments, incorporating such elements as flow straighteners on the turning cascade. Many tests may be performed on such a bed. Typically, such a rig is used to test the reliability of engines, the transient behaviour, ground starting, the control system operation, surge margins and handling characteristics.

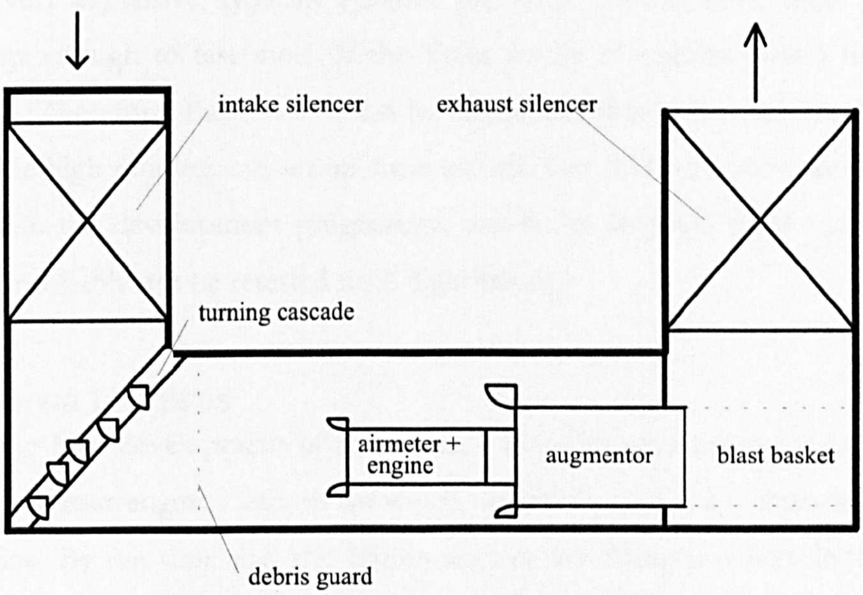

Figure 80 - Sea Level Test Bed [65]

Altitude Test Facility

For some testing, it is necessary to perform the tests at conditions of reduced pressure and sometimes temperature, as encountered at altitude. For these tests, an Altitude Test Facility (ATF) is used, as shown diagrammatically in Figure 81. Such tests include those for windmilling and relight, but also include some handling tests. Windmilling and relight tests cannot be conducted on a standard test bed as they require a pressure drop across the engine to be maintained by the rig. However, some tests may be conducted at ambient temperatures, provided that the comparisons with the performance model include the effects of this temperature difference from that experienced at altitude.

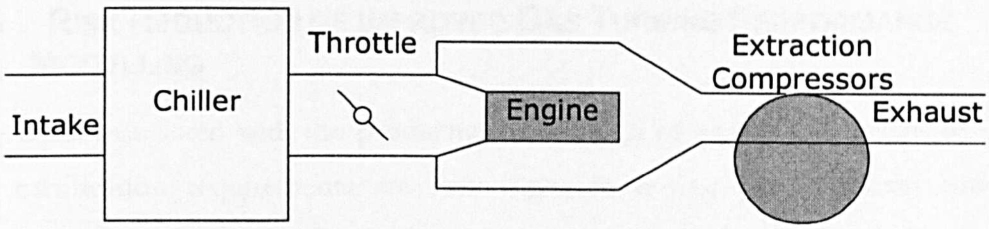

Figure 81 - Altitude Test Facility

Due to the requirement for large airflows and the chiller system, the operation of an ATF is very expensive, typically £20,000 per hour. Furthermore, there is not an ATF large enough to test most of the Trent family of engines within the United Kingdom. Therefore, these tests must be conducted abroad, at additional expense. Due to the high running and set-up costs and the fact that ATF tests are conducted very late in the development programme, any faults detected while running ATF tests will probably not be retested until flight testing.

FLYING TEST BEDS

During the final development of an engine, it is tested on a flying test bed. This is essentially a four-engined aircraft on which one of the engines is replaced with the test engine. By the time that the engine reaches the flying test bed, its behaviour should be well understood. Therefore, the flying test bed is used mainly for tweaking the control systems.

6.4 CERTIFICATION

Before an engine can enter service, it must be certified by the airworthiness authorities. This certification process includes a number of tests to ensure that the engine meets its requirements. These tests include the bird-strike and the fan blade failure cases. However, two of the certification issues which are of interest to this work are the relight envelope and the power offtake availability. Should the engine fail to meet the agreed requirements, it would not be allowed to enter service, a huge risk as it means that the company would not receive income from its sales, may have to pay fines because of penalty clauses and because the company would gain a bad reputation.

6.5 RISK REDUCTION OF IMPROVED GAS TURBINE PERFORMANCE MODELLING

The costs associated with the producing of a design of engine which fails to meet the certification requirements are very high. As engine design moves towards smaller, higher pressure ratio, engine cores powering larger offtake demands, the windmilling and relight criteria become increasingly difficult to meet.

As time increases throughout the project, the commitment of funds rapidly becomes greater. Therefore, potential problems with a design must be spotted early in the project lifetime in order to avoid costly mistakes. With relight and windmilling power offtake availability being two areas in which an engine design may fail to meet the necessary requirements, such a possibility should be investigated as early as possible and then monitored as the engine design becomes more final.

If reasonable sub-idle characteristics can be estimated from the design point performance, approximate prediction of windmilling and relight performance should be possible as early as Stage 1 of a project. Rolls-Royce's policy of incrementing technology changes in the Trent family of engines makes this more of a possibility, as generic, scaled, sub-idle characteristics should produce reasonable estimates of turbomachinery performances. It is important, however, in the scaling of the characteristics, to bear in mind the arguments, presented in this thesis, of considering turbomachinery performance to be a combination of a torque and a pressure loss. Scaling according to these variables will produce different scaled maps than scaling on more traditional variables. The resultant characteristics can then be coupled to the work performed within this project on power offtake systems by Vielma [78] to produce a windmilling model of reasonable reliability.

When the engine design process moves to Stage 2, it is likely that Computational Fluid Dynamics (CFD) models will be created of the turbomachinery components. If these models are run for a number of different inlet Mach numbers for the locked rotor case, as in the work performed by Chambard [11] early in the computational testing of the component design, this can, relatively quickly, facilitate the production of more accurate sub-idle characteristics. Thus, again, there is potential to spot potential problems associated with windmilling and relight earlier in the product development.

The ability to detect potential problems with a design early in the engine development process has enormous benefits in terms of reducing the risk of redesign, a process which is expensive in terms of both the cost of the manpower and testing resources of the redesign itself, and in terms of the penalties associated with the late delivery of a new engine and the poor reputation which will follow as a result of such delays. Due to the nature of the risk, it is difficult to quantify it in terms of cost. However, it is certain that problems at a late stage of the development process would prove to be very expensive indeed.

6.6 COSTS BENEFITS OF IMPROVED GAS TURBINE PERFORMANCE MODELLING

In addition to reducing the risk, improved modelling has clear benefits in terms of development cost reduction. Most of these savings are to be seen at the Altitude Test Facility. At a very high hourly running cost, a relatively small reduction in the test schedule of an engine can produce large cost savings.

A significant proportion of the testing conducted for windmilling and relight in the ATF is associated with the determining of appropriate control system strategies. If the performance model used in the earlier stages of engine development were more reliable than is currently the case, it would be possible to determine more of this control system strategy before reaching the ATF. It is inevitable that some tweaking of the control system would still be required, as no model will ever completely match the engine behaviour. However, large savings in the test schedule should be possible.

Furthermore, during the testing of an engine for relight, the engine is run powered for a time. The fuel is then cut and the engine spools down. The engine is then allowed to free windmill for a considerable time before relight is attempted. This allows the engine to thermally stabilise and reflects reasonably well the situation in which an engine flames out at an altitude above the relight envelope and the pilot then descends to an altitude suitable for attempting a relight. However, if the performance model were more able to predict the effects of this heat transfer, it would be possible to conduct the tests with a shorter time of steady-state windmilling and use the performance model, now calibrated through testing, to predict the behaviour after thermal stabilisation. While this work does not yet satisfy this requirement, further improvements in windmilling and relight modelling could.

6.7 SUMMARY

The design and development process at Rolls-Royce has been outlined. Then, an analysis has been conducted of how improvements in modelling may affect this process, reducing the risk of design problems and reducing the development costs. Recommendations have been made for areas to pursue in the continuation of this work.

It has been demonstrated that the effects of this research could be quite substantial, in terms of reducing design risks and costs, and possibly in minor alterations to engine and aircraft design.

7 The Effort Multiplier of Sharing the Work

7.1 INTRODUCTION

During the course of this doctorate, various M.Sc. and SOCRATES students have worked on the project, broadening its scope and providing more depth to the work. The use of these students has proven to be very useful in this project and the benefits of this are described in this section.

While partaking in the project, the students were enrolled in a variety of courses. The majority came from the Master of Science in Thermal Power, a taught course mainly studying gas turbine engineering, some of whom were using the year in Cranfield as part of their undergraduate degree in their home country. One student was undertaking a Master of Science by research, a full-time research programme. Finally, three students were on six-month placements through the SOCRATES exchange programme from their home universities' undergraduate programmes.

7.2 THE STUDENTS AND THEIR WORK

During the first year, Sog-Kyun Kim [35] developed a low-speed gas turbine performance model using SIMULINK (Section 4.2). In the second year, Cyrus Haghrooyan [25] worked on the effect of fuel temperature changes on the ability of the combustor to ignite during windmilling conditions (Section 5.2.2). In the third year, this work was complemented by Mathieu Kupcic's [38] study of bulk heat soakage in the compressor system and the subsequent effect of combustor air temperature changes on ignition capability (Section 5.2.5). In the final year, Vicent Chulvi [14], Joan Josep Moncholí [56] and David Gattini [24] studied the method of using empirical methods of representing the stage performances of compressors and turbines in order to extrapolate the component characteristics (Section 3.3.4). Further work was also done in the fourth year by Chris Allan, Omar Vielma and Romain Chambard. Chris Allan [5] looked at the effect of combustor heat transfer at light-up and acceleration conditions (Section 5.2.6). Omar Vielma [78] performed an analysis of the starter, gearbox and power offtake systems at the low speeds encountered during starting, windmilling and relight (Section 5.3). Romain Chambard [11] investigated the feasibility of producing a CFD model of compressor behaviour at locked rotor conditions with a view to the interpolation of data and the generation of the sub-idle characteristic (Section 3.3.2.8).

7.3 SYNCHRONISATION OF OBJECTIVES FOR AN EQUITABLE OUTCOME

7.3.1 Introduction

In this section, the differing objectives of those involved in the project are examined, with a view to providing an equitable compromise of these objectives which can satisfy all the objectives as much as possible.

First, the needs and goals of each group of players are investigated. Then, these needs and goals are examined to find ways in which they may be simultaneously fulfilled.

7.3.2 Objectives of the Players

7.3.2.1 Introduction

While the main part of this work has been an Engineering Doctorate project performed for Rolls-Royce plc, it has also seen involvement from a number of other individuals and groups. Thus, the project's main players are: Rolls-Royce plc, who provide some of the funding for the work; EPSRC who provide the main part of the funding; Cranfield University, through whom the work was done and from whom most of the supervision came; Geoff Jones, the author of this thesis and the main researcher on this project; the M.Sc. students; and the SOCRATES exchange students. This section investigates the objectives of each of these groups in the project.

7.3.2.2 Rolls-Royce

The work contained within this thesis was performed for Rolls-Royce plc and was partially funded by them. As the main customer, they had a number of objectives:

- Improve modelling methods for windmilling, relight and starting. This is the basic brief for this project. Through the achievement of this objective, it is hoped that the following objectives may also be addressed.

- Reduce modelling effort, complexity. Currently, the modelling of gas turbines for windmilling and relight is somewhat complicated and thus there are only a limited number of staff who can set up such an analysis. These people, by their nature, are usually heavily in demand, as well as being more expensive than others. Therefore, it is preferable to simplify the process of setting up windmilling and relight synthesis in order to reduce demands on limited human resources. This can be achieved if the methods developed are either automated or can be carried out by following a simple set of instructions.

- Improve personnel capabilities in windmilling and relight modelling. While one aim of such work is to reduce the skills requirement of staff, the problem of limited human resources may also be approached from the opposite side. Through the transfer of knowledge from this project to employees within Rolls-Royce, or through the recruitment of members of the project team, the company can improve its base of knowledge and understanding of windmilling and relight.

- Reduce risk of certification problems and redesign. Should an engine design fail to meet the requirements of the certification process as discussed in Section 6.4, a redesign of the engine may be necessary. This process can put the launch of an engine a long way behind schedule and cost the company huge sums of money. Therefore, it is important that the design is right first time. In order to achieve this, models must be developed which can predict the windmilling and relight performance of an engine before large resources are committed to the design.

- Reduce testing time. As discussed in Section 6.6, engine development testing is an expensive procedure. Therefore, reductions in the time spent testing has enormous cost benefits. Through improved windmilling and relight models, test schedules can be substantially reduced.

- Gain new recruits. Through exposure of Rolls-Royce to students at Cranfield University and through working closely with them, competent new employees with excellent skill sets can potentially be recruited, increasing the company's capability.

7.3.2.3 EPSRC

The Engineering Physical Sciences Research Council has provided the majority of the funding for this work. Their brief is very broad and open, and is outlined below [22]:

- Enhance economic development and commercial competitiveness. Through investing in research in a company which competes in a world market and the majority of whose products are exported, the economy of the UK benefits. This is seen in terms of trade balance and in the lower unemployment brought about through successful companies.

- Stimulate interest in the sciences. Investing in the youth of a nation stimulates more people to join the professions of science and engineering. This benefits the nation as a whole, as more trained scientists and engineers create more competitive industry.

7.3.2.4 Cranfield University

Cranfield University, through the Rolls-Royce University Technology Centre in Gas Turbine Performance Engineering and through other joint research ventures, maintains close links with Rolls-Royce plc and a substantial proportion of their research income is from industry. They provide a number of Masters and doctoral programmes of study as well as a number of short courses.

They have the following requirements for this and similar pieces of work:

- Maintain and increase funding for the UTC. By delivering high quality research to Rolls-Royce, Cranfield University should be able to attract more funding from Rolls-Royce and from other companies more easily.

- Raise profile of university in the world. In presenting research at conferences and in journals, accompanied by the accreditation of a major respected company such as Rolls-Royce, the profile of Cranfield is raised. This then leads to more student applications, more short course attendees and more industry sponsorship.

- Improve competence in the areas of the research. By monitoring the research, staff at Cranfield may improve their own personal competencies. Furthermore, if theses are well-written, they provide a sound base for future researchers to continue the work already achieved.

7.3.2.5 Geoff Jones

Geoff Jones, the author of this thesis, has been performing research as part of an Engineering Doctorate. His main objectives for this research are outlined below:

- Produce a contained piece of work of a sufficient standard for a doctorate. This is the ultimate main goal of the researcher's attendance at Cranfield.

- Utilise the work of the M.Sc. and exchange students to extend the scope of the doctoral research. In order to enhance the value of the doctorate, other researchers were used in addition to Geoff Jones. This resulted in the production of a thesis of broader scope than would have been possible with just the lead researcher.

- Improve knowledge and understanding of engineering. By investigating a subject area which covers many different aspects of engineering and applied physics, the researcher's knowledge and understanding should be enhanced. This is further improved when the work of others can be absorbed.

- Improve people-management skills. By working in a team and managing a number of other researchers in the overall project, the lead researcher's interpersonal skills should be improved.

- Improve employment prospects. By obtaining an engineering doctorate and by publishing papers and thus becoming known to influential members of industry, the employment prospects of the researcher should be enhanced. This should lead to more interesting job offers and more attractive salaries.

7.3.2.6 M.Sc. Students

A number of Master of Science students have been involved in this project. Their aims are described below:

- Produce a contained piece of work of a sufficient standard for a Masters thesis. This is the main goal of the student.

- Improve employment prospects. By developing links with Rolls-Royce and by performing a project which is to be applied in industry, a student may expect to improve their prospects for employment.

- Improve knowledge and understanding of engineering. In the process of doing the project work, a student will have to learn about new aspects of physics and engineering and how to apply them. This will lead to their being more marketable in job seeking.

7.3.2.7 SOCRATES Students

Three SOCRATES exchange students visiting Cranfield University for a period of six months were involved in this project. Their main objectives were:

- Produce a contained piece of work for a SOCRATES project. The requirements for a project on an exchange programme are generally not as stringent as those for an M.Sc. However, the main goal of the programme remains the achievement of writing a project report.

- Improve knowledge of the English language. As the exchange students invariably come from non-anglophone nations, one of the main goals of the students is to improve their knowledge of English.

- Learn about British culture. While on the exchange, a student will want to gain an understanding of life in a country outside their own.

- Improve employment prospects. As with the other student researchers, the exchange students wish to improve their employability through working for a well-known large engineering company.

- Improve knowledge and understanding of engineering. During the course of the project, many new skills should be learnt. These become valuable assets in an engineer's working life.

7.3.3 Finding Synchronised Objectives for a Win-Win Situation

Inevitably, it is not possible to fulfil every objective of every player fully. However, it will usually be possible to fulfil most of the objectives to an acceptable degree. This section investigates the creation of an appropriate compromise.

In many respects, the goals of all the players are naturally aligned. Through their own individual objectives, each of the researchers wish to create a contained piece of work and a well-written report based upon that. This then ensures that the sum of the available knowledge is increased as the work is in progress, and that a means is provided for the transfer of that knowledge in the form of a report.

One area which can potentially cause problems with any piece of work where jobs are delegated is the tendency to do this minimum possible to achieve the objectives.

Given that this is the case, it is important to ensure that the requirements of the project are met with little effort expenditure on the part of the student over that which is required to meet their own objectives. If possible, the situation should be created whereby it is actually easier for the student to fulfil the wider set of objectives than not to. This can be achieved in the important area of knowledge transfer as, during the weekly meetings with the project leader, knowledge will be imparted to the leader, while the student is rewarded for this knowledge transfer by gaining the assistance of the leader in tackling their individual problems.

In those cases where a student's work could be made more useful by expanding it a little in a direction other than the main focus of the sub-project, a way should be found whereby this can be shown to be beneficial to the sub-project also. Examples here include the use of the locked rotor CFD model to investigate the effects of a change in the working fluid and the extension of power offtake modelling to include bearing losses.

7.4 DETERMINING OF AN APPROPRIATE LEVEL OF SUPERVISION

7.4.1 Introduction

In the supervision and management of any work, a balance must be struck between providing enough guidance to provide focus and security and providing the freedom to explore. Given the different mentalities of different people, this balance must vary from one person to the next. However, some basic rules can be outlined.

7.4.2 The Start of the Project

At the start of a project, it is important to give a brief to the student. This brief should be written to be sufficiently well defined to avoid the student investigating areas which are of limited interest to the larger project, while not limiting the work to using one particular method where the student may find a better solution. The degree to which each sub-project was defined was different for the SOCRATES and M.Sc. students, with the project definitions being more detailed and with less scope for investigation for the exchange students, whose time at Cranfield was less and who were at an earlier stage of their academic development.

The students should be given a clear view of where their work fits into the overall scope of the project. This should be done through giving an overview of the whole project and by setting boundaries to their responsibilities. Such an approach provides a good degree of security for the student.

Should further guidance be required, it should be made available. However, in the early stages of a project, it is best to give the minimum direction required for the security of the student, as too much can cut off some initial thinking, leading to poor understanding later and sometimes to fundamental misunderstandings or false assumptions.

7.4.3 During the Project

Once the student has started work on the project, supervision and guidance must be maintained in order to ensure that the work remain focused on the end goal, that the results are presented in such a fashion as to be useful and that the work remains on schedule. This was achieved through the meetings described in Section 7.7. During these meetings, the project leader or the supervisor made an assessment of the above and provided steering where necessary.

Assistance was provided by the project leader where necessary. In particular, some students had some limitations on their computing skills, while others had difficulty with some aspects of the mathematics involved. In these cases, the project leader would give occasional assistance in order to ensure that the work was completed successfully.

Towards the end of the project, at the time of writing up what had been achieved, the project leader was also available to do some proof-reading. Despite taking up a considerable amount of time, this proved to be beneficial in those cases where students were able to provide drafts of their reports in advance as it permitted the clarification of any points of ambiguity or uncertainty. It also ensured that the main part of the thesis had been read before the student had left. In those cases where a draft or final copy had not been received prior to the departure of the student, it sometimes took some time to obtain a copy of the thesis, resulting in a delay to the work's incorporation into the overall project.

7.5 DISTRIBUTION OF WORK

In order to define the sub-projects effectively, a work breakdown structure was used for the overall project. This is shown in Figure 82. The sub-project definitions were then built upon this structure.

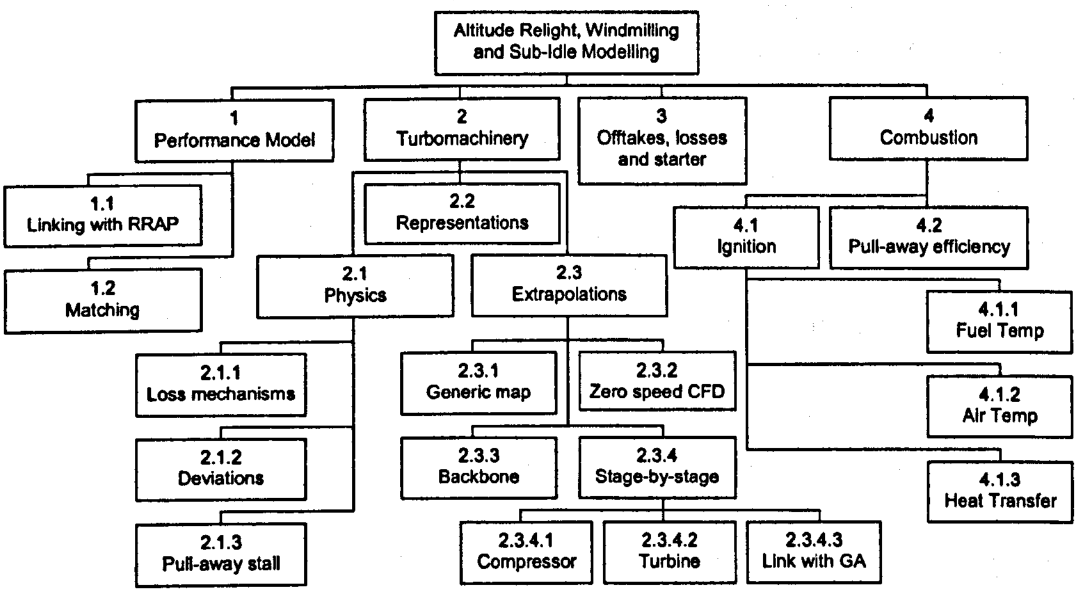

Figure 82 - Work Breakdown Structure for Work Package

Some of the components of the project shown in Figure 82 are dependent on the prior completion of other components. However, this is only the case for a few components. Where such an issue was encountered and work had not already begun on the prerequisite, one of the following was necessary:

- A parallel sub-project which would produce the necessary results quickly
- The project leader performing the necessary work

- The student performing a cut-down version of the required work sufficient to allow the main part of the sub-project to continue

This was encountered on a number of occasions. For instance, for the work of Mathieu Kupcic [38], a knowledge of the combustor inlet temperature variation was needed. This was calculated using a relatively simple method by the student. In another example, Omar Vielma [78] required the framework of a gas turbine performance model in order to write the code for his power offtakes routines. This model was provided by the project leader.

7.6 KNOWLEDGE FLOW

This section describes the flows of knowledge within the project and the means by which these knowledge transfers took place. The main flows are summarised in Figure 83.

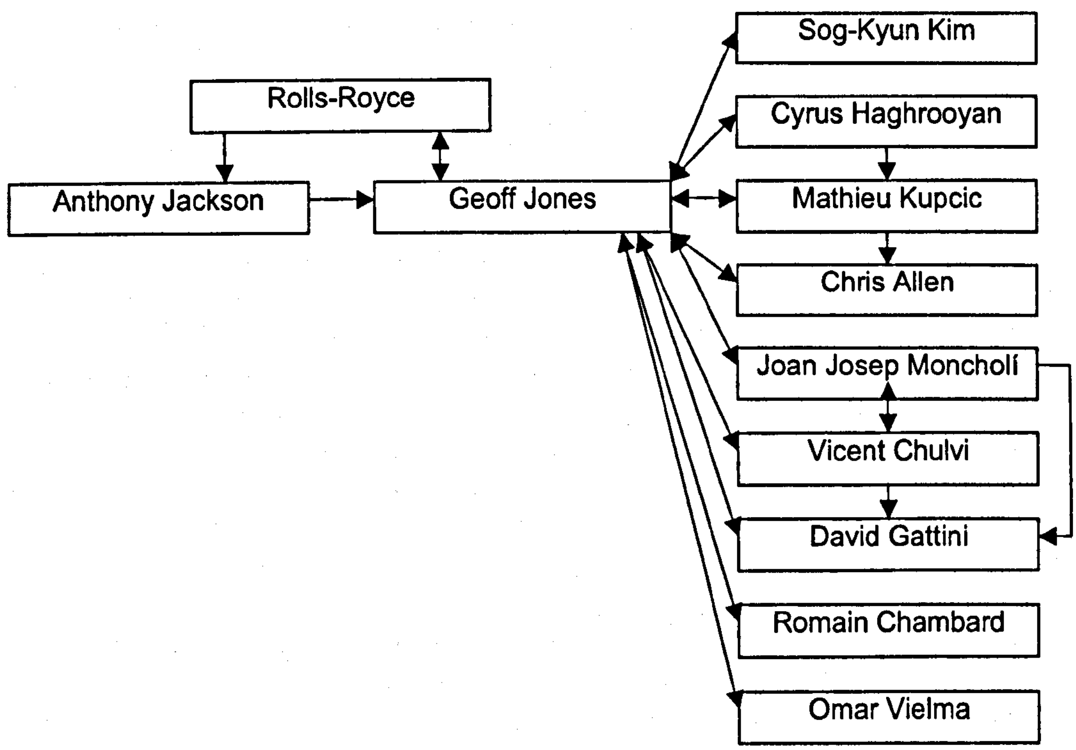

Figure 83 - Main Flows of Knowledge in Project

In order to provide the groundwork for this project, a review was performed by Tony Jackson [31] of the current situation with regard to performance modelling of windmilling and relight within Rolls-Royce.

7.7 REPORTING PROCEDURES AND KNOWLEDGE CAPTURE

7.7.1 Introduction

In order for the work of the project members (students) to be utilised, it must be captured by the project leader (Geoff Jones) and incorporated into the overall project. This must then be presented to the customer (Rolls-Royce). For this to be achieved successfully, the project leader must maintain a close view of the progress of the project members. This is normally achieved through a combination of meetings, reports and presentations.

During the course of the project work of the M.Sc. and exchange students, weekly, informal meetings were held with the project leader. These were then supplemented by monthly meetings with their academic supervisor, Professor Pilidis. Reporting to Rolls-Royce was made via quarterly presentations and an interim report and a thesis.

7.7.2 Weekly Meetings with Project Leader

The meetings between the project members and the project leader took place on a roughly weekly basis. Unless there was significant crossover between the sub-projects, these meetings were on a one-to-one basis.

Being weekly, the meetings were informal. They served a number of purposes:

- Providing a forum to discuss problems which project members had encountered
- Permitting the monitoring of progress
- Allowing transfer of findings to the main project as soon as they became available
- Providing extra incentive to make progress in order to have something to report for the next meeting

An open, relatively informal atmosphere was intentionally maintained for the meetings. This then created an environment in which honesty and openness could be maintained, as without them it would be difficult to obtain a clear view of the state of the work. Thus, if a student requires help, he should be more prepared to ask for it than in a more formal environment, and not put up a pretence of confidence. The informal atmosphere also ensures that the time which the student needs to prepare for the meeting is low and thus the disruption to their work is minimal.

7.7.3 Monthly Meetings with Academic Supervisor

Roughly once a month, meetings were held between the M.Sc. students, the project leader Geoff Jones and the academic supervisor Prof. Pilidis. These meetings were more formal than those between the students and the project leader and were essentially progress reviews. The main purpose was not the transfer of knowledge, but to check that the students' work was on schedule and of sufficient standard and depth to be suitable for a thesis. Thus, the form of the meetings was a small verbal presentation to Prof. Pilidis by each of the M.Sc. students in turn, using visual aids where necessary, followed by the opportunity for discussion, usually led by Prof. Pilidis.

7.7.4 Quarterly Presentations to Rolls-Royce

In order to steer the students' work towards the objectives of the company, formal presentations were made by the students to Rolls-Royce on a roughly quarterly basis. These presentations took the form of a clear statement of objectives and a progress report. Thus, it was possible for representatives of Rolls-Royce to discuss how well the stated objectives fitted in with the company's requirements and whether the work was making good progress and thus represented a good return on investment. The presentations were then followed by the opportunity for discussion.

7.7.5 Formal Reporting

During the first half of the academic year, the M.Sc. students were required to write a short formal report stating the aims of their projects. This could then be reviewed by Rolls-Royce representatives before much time had been spent working on the project, in case it was decided that the objectives needed to be heavily revised.

At the end of their work, the students each produce a thesis. This details the methods which they used and the results of their work, with a view to being able to implement the methods at Rolls-Royce.

7.7.6 Summary

Through a combination of different forms of meeting and reporting, the following needs were met:

- Transfer of knowledge from students to project leader
- Supervision of student progress
- Alignment of project objectives with Rolls-Royce requirements

7.8 DISTRIBUTION OF STUDENT EFFORT OVER TIME

In Figure 84, the distribution of student effort over the course of the project can be seen. Apparent from this chart is the clustering of student effort into the later stages of the work. This was not an intentional situation, but was rather a consequence of the way in which students choose a project on which to work. Masters projects are advertised to students through a student handbook and an introductory session, both at the beginning of the academic year. However, the number of available projects is usually far larger than the number of students, resulting in a rather random distribution across the years.

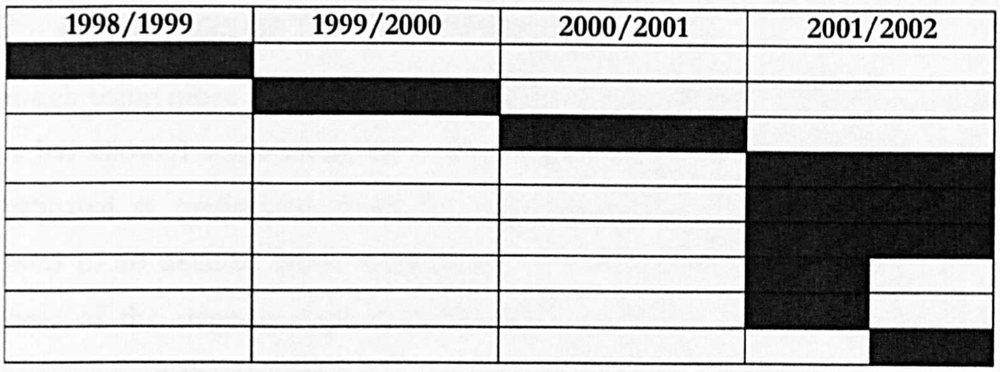

Figure 84 - Distribution of Student Effort over Time

At Cranfield, Master of Science projects finish towards the end of August, while the Engineering Doctorate programme finishes at the beginning of October. Thus, work performed by M.Sc. students during an Eng.D. student's final year is only finished some six weeks before completion of the doctorate. This inevitably reduces the usefulness of the M.Sc. work for the overall project. For this reason, a clustering of effort towards the end of the doctorate is not recommended. It should be noted here that this is not necessarily true when a Ph.D. is being studied, as the Ph.D. does not have the same strict hand-in time as the Eng.D. programme, and thus the work of M.Sc. students may be incorporated more easily. This is not as much of an issue with exchange students, as they normally finish somewhat earlier.

During the first year of the doctoral project, it is usually difficult to utilise the efforts of others effectively, as the principle researcher will normally face a steep learning curve themselves and will thus not be able to give the necessary mentoring to the M.Sc. students. However, a limited number can be used to perform preliminary investigations of some aspects of the overall project, as was done in this project.

The net result of the above is that the optimum distribution of supplementary effort is perhaps one student in the first year, then larger numbers in the second and third years, with few in the final year. The work of exchange students can be utilised more easily than that of M.Sc. students in the final year, but researchers should be aware of the extra effort required to mentor students at earlier stages of their academic development.

7.9 ADVANTAGE OF THE EFFORT MULTIPLIER

Through using more manpower on a project, the project's scope may be widened. This has allowed some areas, such as the heat transfer during spool-down, to be investigated as preliminary work for future researchers to continue. It has also allowed more detailed work to be done on some areas, such as a more refined analysis of the stage-by-stage semi-empirical method of creating an extrapolated turbomachinery characteristic.

A further advantage, which should not be underestimated though apparently small, is the psychological effect which it has on the project leader. Performed alone, a project of four years is an experience of solitude, which can sometimes mean poor levels of motivation. However, the presence of other members of the project, and particularly their dependence on the results being produced, provides intermediate objectives, making the overall project an easier affair.

The building of a team for the performing of a project also brings different views. The different opinions and outlooks of different people can contribute greatly to the overall project, as it helps provide an overview which would not be possible by one person alone.

7.10 SUMMARY

This section described the work which was performed by the students, how the work was allocated, how it was supervised and how the contributions of each student were incorporated into the overall work.

It has been shown that Masters and exchange students can provide valuable contributions to doctoral research, broadening the scope of a project and providing a better overview of the overall work.

8 Discussion

8.1 INTRODUCTION

This section investigates the implications of the work performed during this thesis, a critique of the work and the scope for future research.

8.2 CRITIQUE AND LIMITATIONS OF WORK

8.2.1 Overview

This section describes the problems which have been encountered with the work. The various aspects which have been investigated are discussed in turn.

8.2.2 Turbomachinery Characteristic Representations

The turbomachinery characteristic representation technique recommended in Section 3.2 is based upon sound physics and trusted correlations. Partial validation of the technique has been performed through the CFD model described in Section 3.3.2.8 at locked rotor conditions. Further validation should ideally be performed. However, this is not practical through experiment.

8.2.3 Turbomachinery Characteristic Extrapolations

The extrapolation techniques need further development in order to be used in the generation of sub-idle turbomachinery characteristics. The method which should be employed to develop the techniques has been outlined.

Validation of the extrapolations is necessary and experimental methods of providing this validation have been outlined.

8.2.4 Combustion Modelling

The methods used for predicting the changes to ignition loops have shown results which are either more extreme than or contrary to what might be expected. Nevertheless, they show that this is an area for concern and therefore should be investigated further. It is not recommended that the results of this section of the work be applied directly, but rather that they demonstrate that more work is required.

The work conducted on the heat transfer in the combustor contained a number of assumptions and should thus be regarded as a preliminary study.

8.2.5 Modelling of Power Offtakes

The power offtake models are believed to be reasonably reliable and accurate. However, they require much specific information on the design of the accessories.

8.2.6 Solver Scheme and Programming Structure

The solver has been shown to be reasonably robust, although it does still fail to achieve a solution for a limited range of points. The speed of the solver is a little slow at present, although this can be improved relatively easily. At present, the spool speed part of the solver is using many steps in the initial line search; this can be reduced. Furthermore, the use of the betas to seed the flow matching could be fine-tuned.

8.3 RECOMMENDATIONS FOR FURTHER WORK

8.3.1 Main Areas for Further Development

This project has largely concentrated on the areas of turbomachinery characteristics and a solver mechanism for steady-state performance. However, the work must be developed to model more completely the steady-state and transient performance of more complicated designs. Furthermore, the methods should be tested against experimental data where available to verify their accuracy. This section explores what work is required.

8.3.2 Extension to Real Engine Designs

The techniques for turbomachinery characteristic representations and solver logic have been tested on a single-spool turbojet engine. However, the robustness of the numerical system has not been tested on a multi-spool engine. Also, the accuracy of the results has not been validated as experimental data are only available for recent, more complicated, engine designs.

8.3.3 Component Testing

8.3.3.1 Turbomachinery

In Section 3.4.3, the testing of a compressor for validating the turbomachinery characteristic extrapolation methods was proposed. This testing should ideally be conducted and the results of extrapolations compared with the experimental data.

8.3.3.2 Combustion

Combustion testing should be used to properly quantify the observations made by Haghrooyan and Kupcic. This must take the form of tests on the spray characteristics of the injectors under varying fuel and air conditions, and separate tests on ignition performance under such conditions. As stability loops are expected to behave in a similar manner, the same test could probably be used to investigate this.

8.3.4 Component Modelling

8.3.4.1 Turbomachinery Modelling

OVERVIEW

The methods for representing the turbomachinery in the gas turbine performance model are believed to be reliable. Therefore, no further work is required in this respect.

The extrapolation of turbomachinery characteristics requires a little more work. This is in respect of the further development of the locked rotor computational fluid dynamics analysis and the stage-by-stage method of characteristic extrapolation.

LOCKED ROTOR CFD

As noted in Section 3.3.2.8, the CFD simulation of a locked rotor condition entered a cyclical mode when flow separation was encountered. Therefore, the feasibility of conducting the simulations with unsteady flow should be investigated.

The simulations run by Chambard were made using a mesh generated at Cranfield and were conducted in two dimensions. However, at Rolls-Royce, the development of a compressor design will include the conducting of three-dimensional computational simulations. Therefore, it would be beneficial to see if the meshes thus generated could be used for performing an analysis of the locked rotor behaviour. If the simulation could produce results within a reasonable computational time, this would save much time and effort in grid generation.

Finally, while comparison with experimental results was conducted as far as possible, full data were not available and so it would be beneficial to compare the CFD results with experimental data if and when these become available.

STAGE-BY-STAGE EXTRAPOLATION

The stage-by-stage extrapolation technique described in Section 3.3.4 has been partially developed and shows promise for generating turbomachinery characteristics. However, the method requires more development. Specifically, weightings must be found for the relative importance of the pressure loss and torque. Then, a technique of evaluating the error must be developed and the means of minimising the error must be incorporated.

8.3.4.2 Combustor Modelling

During the course of this project, some aspects of combustor modelling have been examined. However, these techniques require some validation and further development. Furthermore, models must be developed to determine the stability of the combustion during pull-away and the combustion efficiency. The models of spray characteristics found in this work will be of use here.

8.3.4.3 Power Offtake Modelling

Vielma produced a model for the power offtake systems of the gas turbine. The model developed is thought to be realistic. However, it requires knowledge of a considerable amount of information on the offtake systems. While this information may be readily available during the later stages of engine development, it is not when the engine is at the preliminary design stage. Nevertheless, models of windmilling and relight performance need to be developed at these early stages and therefore it would be beneficial to produce a simplified version of the offtakes model, using approximate values of parameters.

8.3.4.4 Modelling of Other Components

Throughout this project, only the turbomachinery, combustion system and power offtakes have been considered in any detail. The influence of the intakes, mixers, nozzles and internal flows have been considered small. However, they should be considered in future work.

8.3.5 Extension to Transient Modelling

So far, the work has concentrated on the steady-state windmilling only, although care has been taken to consider transient modelling issues also. While the steady-state model alone is sufficient to determine many of the important aspects of windmilling and relight modelling, it is not capable of producing an analysis of full relight performance, particularly in the case of quick relights. Therefore, the work should be extended to cover transient modelling.

The solver scheme used can be modified relatively easily to transient analysis. Furthermore, the turbomachinery characteristic representation methods were developed with the requirement of transient modelling in mind. However, methods have not been developed within the scope of this work to look at the effect of heat soakage on turbomachinery performance or the combustor stability and efficiency during pull-away. Therefore, further work is necessary on these two issues.

Some work has been done on the heat transfer within the turbomachinery and the combustor as a precursor to work on transient modelling. This work should be developed further.

8.4 IMPLICATIONS

8.4.1 Testing Changes

As the models of the windmilling behaviour of engines improves as a result of this work, the time required for testing can be reduced, with large cost savings, as discussed in 6.6.

8.4.2 Engine Modifications

This work has highlighted a number of issues in windmilling and relight which may be improved through modifications to engine design.

One issue which has been noted is the influence of fuel temperature on the ignition performance of an engine. This temperature can be raised by a number of means, including cross-bleeding fuel from a running engine (Figure 66), electrical heating of the fuel or bypassing the oil cooler as soon as a flame-out is detected.

The influence of the oil temperature on the windmilling performance was also observed. Again, the windmilling performance could be improved through switching the oil cooler off as soon as a flame-out occurs.

9 Conclusions

9.1 INTRODUCTION

In this section, the main conclusions of the work are presented. These conclusions are presented in sections relating to the areas of the research. A summary of the key achievements of the work is then presented.

9.2 TURBOMACHINERY CHARACTERISTIC REPRESENTATION

The theory underpinning the use of turbomachinery characteristics was examined. Then, various techniques of representing the performance of the components were investigated. Resulting from this investigation, a modified pressure loss parameter (Eq. 5) was derived. Both the compressor and turbine characteristics were defined using the same parameters, with beta and non-dimensional speed for the operating point definition. The full set of mapping parameters are shown in Table 2 to Table 5.

9.3 TURBOMACHINERY CHARACTERISTIC EXTRAPOLATION

The need to extrapolate turbomachinery characteristics for sub-idle performance modelling has been explored and a number of methods for executing the extrapolation have been investigated. It was found that the use of a zero-speed characteristic could aid extrapolation, and that a sufficiently accurate locked rotor map could be obtained through CFD analysis. A new technique for extrapolation was investigated. This was a semi-empirical method based on a stage-by-stage analysis.

9.4 COMBUSTOR MODELLING

Through the work of M.Sc. students managed by the author of this thesis, the effects of fuel temperature changes, combustor inlet air temperature changes and combustor heat transfer on the performance of the engine were studied.

9.5 MODELLING OF OTHER COMPONENTS

An M.Sc. student managed by the author of this thesis produced a model of the power offtake requirement of an engine. A flow function was developed for the nozzle inlet of the engine, in order to produce a smooth error function for the engine performance simulation. Other components of the engine were briefly considered, though work remains to be done on them.

9.6 WINDMILLING PERFORMANCE SYNTHESIS

Following from the work on the turbomachinery component representation methods, a strategy for solving the system of relationships which govern the operation of gas turbines was developed. This consisted of a nested iteration. The inner loop of the iteration contained flow matching. This matching was performed on the combustor outlet and turbine inlet and on the turbine outlet and nozzle inlet, varying guesses of compressor beta and turbine beta through the use of the RRAP multi-dimensional solver. The outer loop contained torque matching. The non-dimensional torque was balanced between the compressor, turbine and power offtakes by varying the compressor non-dimensional speed. The iteration scheme worked well, providing a solution in nearly all cases.

9.7 SUMMARY

The research conducted in the course of this project and presented in this thesis has led to the improvement of understanding and analytical techniques in the following areas:

- Analysis of various techniques of representing the characteristic maps of compressors and turbines at windmilling conditions and throughout the engine operational envelope has been performed.
- A technique of representing the turbomachinery components was devised using a new parameter of modified pressure loss and a beta variable for the turbine.
- Methods of predicting sub-idle performance of compressors and turbines were assessed
- New techniques for the prediction of sub-idle turbomachinery characteristics have been proposed:
    - Use of computational fluid dynamics to define the locked rotor characteristic
    - A stage-by-stage semi-empirical technique for the prediction of sub-idle turbomachinery performance from above-idle data
- A nested solver scheme has been used for the solution of steady-state performance modelling, using handles of fuel flow rate, altitude, flight speed and relative humidity, guess variables of compressor beta, turbine beta and compressor non-dimensional speed and error variables of combustor exit - turbine inlet non-dimensional mass flow rate, turbine outlet - nozzle inlet non-dimensional mass flow rate and shaft non-dimensional torque balance.
- Experimental methods of investigating the sub-idle performance of turbomachinery have been proposed.
- The coupling of a gas turbine performance simulation to a spreadsheet package has been carried out to facilitate easy post-simulation analysis while using fast compiled code to perform the main gas turbine simulation calculations.
- A review of the design and development process at Rolls-Royce has been conducted with a focus on the role of performance modelling. The potential benefits of improved windmilling modelling have been highlighted.

10 References

[1] *Estimation of windmilling drag and airflow of turbo-jet and turbo-fan engines.* ESDU-81009, 1981.

[2] Adamopoulos, D. *Handling and transient operation of a military turbofan.* Cranfield University, MSc thesis, 1996.

[3] AGARD-AR-332, 1995, *Recommended Practices for the Assessment of the Effects of Atmospheric Water Ingestion on the Performance and Operability of Gas Turbine Engines.* Sept. 1995, ISBN 92-836-1022-9.

[4] Agrawal, R. K. and Yunis, M. A generalized mathematical model to estimate gas turbine starting characteristics. ASME, 81-GT-202. IN: *Transactions of the ASME, Journal of Engineering for Power,* vol. 104, no. 1, 1982, 194-201.

[5] Allan, C. MSc Thesis, Cranfield University, August 2002.

[6] Ballal, D. R. and Lefebvre, A. H. *Basic ignition research related to altitude relight problems.* IN: *Gas turbine combustor design problems.* Washington D.C.: Hemisphere Publishing Corp., 1980, 189-201.

[7] Bauerfeind, K. A new method for the determination of transient jet engine performance based on the nonstationary characteristics of the components. IN: AGARD, *Advanced components for turbojet engines,* CP-34, 1968.

[8] Braig, W., Schulte, H. and Riegler, C. Comparative analysis of the windmilling performance of turbojet and turbofan engines. *Journal of Propulsion and Power,* vol. 15, no. 2, 1999, 326-333.

[9] Brewster, H. M. and Kush, A. *Development of a turbojet altitude windmill starting correlation.* Naval Air Turbine Test Station, ATL-TN-3, 1958.

[10] Broyden, C. G. Quasi-Newton methods and their application to function minimization, *Mathematics of Computation,* 21, 1967, 568-581.

[11] Chambard, R. *Study of a Compressor during Windmilling Using Computational Fluid Dynamics.* MSc Thesis, Cranfield University, August 2002.

[12] Chappell, M. A. and McLaughlin, P. W. An approach to modeling continuous turbine engine operation from startup to shutdown. AIAA, 91-2373. IN: AIAA, SAE, ASME and ASEE, Joint Propulsion Conference, 27th, 24-26 June 1991, Sacramento, CA. (Also: Journal of Propulsion and Power, Vol. 9, No. 3, May-June 1993, pp. 466-471.)

[13] Choi, M.-S., Kang, I.-S. and Hong, Y.-S. Analysis of windmilling characteristics for a twin-spool turbofan engine. ASME, 97-AA-113. IN: *ASME Asia '97 Congress & Exhibition*, 30 September-2 October 1997, Singapore.

[14] Chulvi, V. *Turbine Characteristics for the Altitude Relight and Windmilling Problem.* Internal Report, Cranfield University, 2002.

[15] Clark, J. and Green, J. Modeling gas turbine combustor performance under transient conditions. AIAA, 90-2161. IN: *AIAA, SAE, ASME and ASEE, Joint Propulsion Conference, 26th*, 16-18 July 1990, Orlando, FL, USA.

[16] Compaq. *Compaq Visual Fortran 6.5 Programmer's Guide.* Compaq Computer Corporation, 2000.

[17] Converse, G. L. and Giffen, R. G., 1984, *Representation of Compressor Fans and Turbines – Volume 1 CMGEN User's Manual.* NASA-CR-174645.

[18] Crainic, C., Harvey, R. and Thompson, A. Real time thermodynamic transient model for three spool turboprop engine. ASME, 97-GT-223. IN: *ASME International gas turbine and aeroengine congress and exhibition*, 2-5 June 1997, Orlando, FL., USA.

[19] Curlett, Brian P. and Felder, James L. *Object-oriented approach for gas turbine engine simulation.* NASA, TM 106970, 1995.

[20] Curnock, Barry. Engine starting and stopping. IN: *VKI, Gas turbine engine transient behaviour.* VKI/LS-1993-06, 1993.

[21] Curnock, Barry. Gas turbine engine steady-state behaviour. IN: *VKI, Gas turbine engine transient behaviour.* VKI/LS-1993-06, 1993.

[22] EPSRC website, *http://www.epsrc.ac.uk/* 2002.

[23] Ganji, Ahmed R., Khadem, M. and Wilcutts, M.. An integrated software for gas turbine simulation. AIAA, 96-2588. IN: *AIAA, ASME, SAE and ASEE, Joint Propulsion Conference, 32nd*. 1-3 July 1996, Lake Buena Vista, Fl., USA.

[24] Gattini, D. *Axial Compressor Characteristic Extrapolation.* Internal Report, Cranfield University, 2002.

[25] Haghrooyan, C. *Combustor Modelling For Ignition Under Windmilling Conditions*, MSc Thesis, Cranfield University, August 2000

[26] Hatch, James E. *Comparison of experimentally and analytically determined windmilling characteristics of a compressor with low over-all pressure ratio.* NACA, RM-E57L12a, 1958.

[27] Horlock, J. H. *Axial flow compressors, fluid mechanics and thermodynamics.* London: Butterworths Scientific Publications, 1958.

[28] Howse, M. G. J. W. *From the Design to Certification of Civil Aero Gas Turbines.* Lecture at Cranfield University, 17th November 1998.

[29] Ibrahim, E.-B. *The predicted transient behaviour of industrial gas turbines.* M.Phil. thesis, Cranfield Institute of Technology, 1991.

[30] Ibrahim, I. B. *The Study of Combustion Efficiency Effect to Altitude Relight Performance of a J85-13 Single Spool Turbojet*, MSc Thesis, Cranfield University, 1995

[31] Jackson, A. J. B. *Aero Gas Turbine Windmilling, Relighting and Pull-Away: a Position Paper*, Rolls-Royce University Technology Centres in Performance Engineering and Combustion, Cranfield University, 20 November 1998, AJCR002. Rolls-Royce Proprietary Data.

[32] Jane's Information Group Limited. *Jane's Aero Engines.* Ed: Bill Gunston. FRAeS. United Kingdom. 2001.

[33] Jones, G. B., Pilidis, P. and Curnock, B. S., Compressor characteristics in gas turbine performance modelling. ASME, 2001-GT-0384. IN: *Proceedings of the 2001 International Gas Turbine and Aeroengine Congress & Exhibition*, 4-7 June 2001, New Orleans, USA.

[34] Jones, G. B., Pilidis, P. and Curnock, B. S., Extrapolation of Compressor Characteristics to the Low-Speed Region for Sub-Idle Performance Modelling. ASME, GT-2002-30649. IN: *Proceedings of the 2002 International Gas Turbine and Aeroengine Congress & Exhibition*, 3-6 June 2002, Amsterdam, Netherlands.

[35] Kim, S.-K. *Gas Turbine Dynamic Simulation Using SIMULINK.* MSc Thesis, Cranfield University, August 1999.

[36] Kim, S.-K., Pilidis, P. and Yin, J. Gas Turbine Dynamic Simulation Using SIMULINK. SAE 2000-PSC-21. IN: *Power Systems Conference*, SAE, 31 October - 2 November 2000, San Diego, California, USA.

[37] Kuo, Kenneth K. *Principles of combustion.* New York: John Wiley & Sons, 1986

[38] Kupcic, M. *Combustor Performance Modelling During Altitude Relight.* MSc Thesis, Cranfield University, August 2001.

[39] Kurzke, J. and Riegler, C. A new compressor map scaling procedure for preliminary conceptional design of gas turbines. IN: *ASME Turbo Expo 2000.* 8-11 May 2000, Munich, Germany.

[40] Kurzke, Joachim. How to get component maps for aircraft gas turbine performance calculations. ASME, 96-GT-164. IN: *Proceedings of the 1996 International Gas Turbine and Aeroengine Congress & Exhibition*, 10-13 June 1996, Birmingham, UK.

[41] Lakshminarayana, B. Methods of predicting the tip clearance effects in axial flow turbomachinery. ASME, 69-WA/FE-26. IN: *ASME Journal of Basic Engineering*, vol. 92, September 1970, 467-482.

[42] Lefebvre, A. H. Current problems and trends in gas turbine combustion. IN: *1st Intern. Symp. on Air Breathing Engines*, 19-23 June 1972, Marseille, France.

[43] Lefebvre, A. H. Fuel Effects on Gas Turbine Combustion-Ignition, Stability and Combustion Efficiency. IN: *Journal of Engineering for Gas Turbines and Power*, Jan 1985, Vol. 107, Pages 24-37.

[44] Lefebvre, A. H. *Gas turbine combustion*, New York: McGraw Hill, 1983.

[45] Lefebvre, A. H. *Theoretical Aspects of Aero-Engine Altitude Relighting Performance.* Propulsion Department, College of Aeronautics, Cranfield Institute of Technology, COA Memo No. 152, R71818, 1968.

[46] Lim, S-K, Roh, T.-S., Hong, Y.-S., Choi, M.-S., Lim, J.-S. Study of Windmilling Characteristics of Twin-Spool Turbo-Fan Engines. AIAA 2002-0376. IN: *40th AIAA Aerospace Sciences Meeting & Exhibit*, 14-17 January 2002, Reno, Nevada, USA.

[47] Longwell, J. P. and Weiss, M. A. High temperature reaction rates in hydrocarbon combustion. *Journal of Industrial and Engineering Chemistry*, Vol. 47, No 8, 1634-1643.

[48] MacCallum, N. R. L., "Axial Compressor Characteristics During Transients," *AGARD CP 324*, Neuilly-sur-Seine, France, 1982.

[49] MacCallum, N. R. L. Thermal influences in gas turbine transients, effects of changes in compressor characteristics, ASME 79-GT-143, 1979.

[50] MacIsaac, B. D. and Saravanamuttoo, H. I. H. Aerothermodynamic factors covering the response rate of gas turbines, Paper 3. IN: AGARD, *Power plant controls for aero-gas turbine engines*, CP-151, 1974.

[51] Mathioudakis, K., Aretakis, N. and Tsalavoutas, A. Increasing Diagnostic Effectiveness by Inclusion of Fuel Composition and Water Injection Effects. ASME, GT-2002- 30032. IN: *Proceedings of the 2002 International Gas Turbine and Aeroengine Congress & Exhibition*, 3-6 June 2002, Amsterdam, The Netherlands.

[52] Mathioudakis, K. and Tsalavoutas, A., Uncertainty Reduction in Gas Turbine Performance Diagnostics by Accounting for Humidity Effects. ASME, 2001-GT-0010. IN: *Proceedings of the 2001 International Gas Turbine and Aeroengine Congress & Exhibition*, 4-7 June 2001, New Orleans, USA.

[53] McLaughlin, Peter. Nonlinear solvers. IN: *VKI, Gas turbine engine transient behaviour.* VKI/LS-1993-06, 1993.

[54] McLaughlin, Peter. Transient engine simulation. IN: *VKI, Gas turbine engine transient behaviour.* VKI/LS-1993-06, 1993.

[55] Michaelides, M. *Gas turbine power off take optimisation.* Cranfield Institute of Technology, MSc thesis, 1993.

[56] Moncholí, J. J. *Compressor Characteristics for the Altitude Relight and Windmilling Problem.* Internal Report, Cranfield University, 2002.

[57] Naylor, P. H. *Effects of heat soakage on gas turbine performance.* M.Sc. thesis, Cranfield University, August 2000.

[58] Oberursel. *Windmilling Relight Task Force Report.* BMW Rolls Royce GmbH, Technical Report No. E-TR534/97-(IR)-ISS00.

[59] Onions, R. A. and Foss A. M. *Improvements in dynamic simulation of gas turbines,* AGARD, CPP-324, 1982.

[60] Owen, A. K., Daugherty, A., Garrard, D., Reynolds, H. C. and Wright, R. D. A parametric study of an axial-centrifugal gas turbine engine using a one-dimensional dynamic engine model and comparisons to experimental results. I – Model development and facility description. ASME, 98-GT-470. IN: ASME, *International Gas Turbine & Aeroengine Congress & Exhibition, 43rd*, 2-5 June 1998, Stockholm, Sweden.

[61] Owen, A. K., Daugherty, A., Garrard, D., Reynolds, H. C. and Wright, R. D. A parametric study of an axial-centrifugal gas turbine engine using a one-dimensional dynamic engine model and comparisons to experimental results. II – Simulation calibration and trade-off study. ASME, 98-GT-471. IN: ASME, *International Gas Turbine & Aeroengine Congress & Exhibition, 43rd*, 2-5 June 1998, Stockholm, Sweden.

[62] Pilidis, Pericles. *Digital simulation of gas turbine transient performance.* Ph.D. Thesis, University of Glasgow, 1983.

[63] Pilidis, P. and MacCallum, N. R. L. *Models for predicting tip clearance changes in gas turbines,* AGARD, CPP-324, 1982.

[64] Pollard, D. and Gostelow, J. P. Some experimental at low speed on compressor cascades. ASME, *J. Engng. Pwr. Trans.* 89 (July), 1967, 427-36.

[65] Puente Perez, C. *Modelling of the airflow within an Engine Test Bed.* M.Sc. thesis, Cranfield University, August 1999.

[66] Riegler, C., Bauer, M. and Kurzke, J., 2000, "Some Aspects of Modelling Compressor Behavior in Gas Turbine Performance Calculations," ASME 2000-GT-574, presented at *Turbo Expo 2000* in Munich, Germany, 8-11 May 2000.

[67] Rolls-Royce. *CUG BS29.* Derby: Rolls-Royce plc, 2002.

[68] Rolls-Royce. *The Jet Engine.* 4th Ed. Derby: Rolls-Royce plc, 1986.

[69] Saravanamuttoo, H. I. H. and Fawke, A. J. *Simulation of Gas Turbine Dynamic Performance.* ASME, 70-GT-23, 1970.

[70] Saravanamuttoo, H. I. H. and MacIsaac, B. D. An overview of engine dynamic response and mathematical modeling concepts, Paper 25. IN: AGARD, *Engine handling*, CPP-324, 1982.

[71] Schobeiri, T., Attia, M. and Lippke, C. Nonlinear dynamic simulation of single- and multi-spool core engines. AIAA, 93-2580. IN: *AIAA, SAE, ASME and ASEE, Joint Propulsion Conference and Exhibit, 29th.* 28-30 June 1993, Monterey, CA. (Also: ASME, *Journal of Propulsion and Power*, Vol. 109, No. 6, Nov.-Dec. 1994, 855-862.)

[72] Seldner, K., Mihaloew, J. R. and Blaha, R. J. *Generalized simulation technique for turbojet engine system analysis*, NASA, TN D-6610, 1972.

[73] Singh, Riti. Combustion in the gas turbine. Part 3: Fuel injection, ignition and stability. CIT/SME/VKI/RS/4. IN: VKI, *Gas turbine combustion*, vol. 1, 1990.

[74] Sobolewski, A. E. and Farley, J. M. *Steady-state engine windmilling and engine speed decay characteristics of an axial-flow turbojet engine.* NACA-RM-E51I06, 1951.

[75] Sosounov, V. A., Tskhovrebov, M. M., Solonin, V. I. and Palikn, V. A. The study of experimental turboramjets. AIAA, 92-3720. IN: *AIAA, SAE, ASME and ASEE Joint Propulsion Conference and Exhibit, 28th*, 6-8 July 1992, Nashville, TN, USA.

[76] Stow, P. *Computational Fluid Dynamics Engineering Simulation.* Lecture, Cranfield University, 2002.

[77] Thomson, B. Basic transient effects of aero gas turbines, Paper 2. IN: AGARD, *Powerplant controls for aero-gas turbine engines*, CP-151, 1975.

[78] Vielma, O. *Windmilling and Altitude Relight: power offtakes limitations during ground starting.* MSc thesis, Cranfield University, August 2002.

[79] Vincent, K. R., Huntley, S. C. and Wilsted, H. D. *Comparison of locked-rotor and windmilling drag characteristics of an axial-flow compressor type turbojet engine.* NACA-RM-E51K15, 1952.

[80] Walker, Curtis L. and Fenn, David B. *Investigation of power extraction characteristics and braking requirements of a windmilling turbojet engine.* NACA-RM-E52D30, 1952.

[81] Wallner, Lewis E. and Welne, Henry J. *Turbine-propeller engine performance in windmilling condition.* NACA, RM-E51J23, 1951.

[82] Walsh, Philip P. and Fletcher, Paul, *Gas Turbine Performance*, Oxford: Blackwell Science, 1998.

[83] Wilsted, H. D. and Armstrong, J.C. *Effect of fuel volatility on altitude starting limits of a turbojet engine.* NACA, RM-E50G10, 1950.

[84] Yan, De-You. *Modelling the axial-flow compressors for calculating starting of turbojet engines.* ASME 85-IGT-97. IN: ASME, Chinese Society of Aeronautics and Astronautics, and China National Aero-Technology Import and Export Corp., *Beijing International Gas Turbine Symposium and Exposition*, 1-7 September 1985, Beijing, People's Republic of China.

[85] Yan, De-You and Mai, Zhong-Fan. *A Dynamic Model of Turbojet in Starting at High Altitude*, AIAA 83-7045, Sixth International Symposium on Air Breathing Engines, 1983.

[86] Yin, J., Hales, R., Pilidis, P. and Curnock, B. High-bypass turbofan model using a fan radial-profile performance map. IN: *Journal of Aircraft Design*, Vol.4, Issue 2-3, pp 115-126.

[87] Yin, J., Hales, R., Pilidis, P., Curnock, B. and Meads, R. Transient Performance for High-Bypass Ratio Turbofan Models. IN: *Aircraft Engineering & Aerospace Technology: International Journal*, Vol. 72, No. 6, 2000.

[88] Yin, J., Hales, R., Pilidis, P. and Curnock, B. 2-Shaft High-Bypass Ratio Turbofan Performance Calculation Using a New 2-D Fan Model. IN: AIAA, *37th AIAA/SME/SAE/ASEE Joint Propulsion Conference*, 8-11 July 2001, Salt Lake City, Utah, USA.

11 Bibliography

[1] Estimation of drag due to inoperative turbo-jet and turbo-fan engines using data item nos. 81009 and 84004. ESDU-84005, 1984.

[2] *Estimation of spillage drag for a wide range of axisymmetric intakes at M 1*. ESDU-84004, 1984.

[3] Andreadis, D. *Altitude ignition/lean decel. study, final report.* Pratt and Whitney Aircraft, West Palm Beach, FL. Government Products Div., AD-A163052, 1985.

[4] Attia, M. S. and Schobeiri, M. T. A new method for the prediction of compressor performance maps using one-dimensional row-by-row analysis. ASME, 95-GT-434. IN: *ASME International Gas Turbine and Aeroengine Congress and Exposition*, 5-8 June 1995, Houston, Texas, USA.

[5] Balakrishan, S. R. and Santhakumar, S. Fuzzy modelling considerations in an aero gas turbine engine start cycle. *Fuzzy sets and systems*, vol. 78, no. 1, 1996, 1-4.

[6] Ballal, D. R. *Axial flow compressor stage stacking techniques*, Ph.D. Thesis, Cranfield College of Technology, 1968.

[7] Borgon, J. (Translated by Kanner (Leo) Associates, Redwood City, Calif.) *Windmilling of the rotor of a turbojet engine with an axial-flow compressor under flight conditions.* NASA, TM-77087, 1983.

[8] Bragagnolo, N., Ferratti, A. and Vananzetti, M. Initial engine relight test on Eurofighter 2000. IN: *1997 report to the aerospace profession: forty first symposium proceedings*, 1st September 1997, 211-231.

[9] Choi, M.-S., Lim J. S. and Hong Y. S. Practical method for predicting the windmilling characteristics of simple turbo jet engines. ASME, 96-TA-60. IN: *ASME Turbo Asia Conference*, 5-7 November 1996, Jakarta, Indonesia.

[10] Davis, M. W., Owen, A. K., O'Brien, W. F. and Cousins, W. T. Joint dynamic airbreathing propulsion simulations partnerships (JDAPS). ASME, 95-GT-279. IN: *International Gas Turbine and Aeroengine Congress and Exposition*, 5-8 June 1995, Houston, Texas, USA.

[11] Eckardt, D. and Brines, G. L. Technology readiness for advanced ducted engines. AIAA, 89-2479. IN: *AIAA, ASME, SAE and ASEE, Joint Propulsion Conference, 25th*. 10-13 July 1989, Monterey, CA.

[12] Garcia Avellaneda, A. *Overall compressor performance prediction from stage characteristics for gas turbine performance simulation*, MSc Thesis, Cranfield University, August 1999.

[13] Gastebois, P. Modelisation des foyers de turboreacteurs: point de vue d'un motoriste (Modelling the combustion chamber of a gas turbine: a mechanics viewpoint). IN: AGARD, *Combustor Modelling*, CP-275. 1980.

[14] Golloday, Richard L. and Bloomer, Harry E. *Investigation of altitude starting and acceleration characteristics of J47 turbojet engine.* NACA, RM-E50G07, 1951.

[15] Goutte, B. *Improved Core Compressor Modelling*, MSc Thesis, Cranfield University, August 1999.

[16] Griffiths, J. F. and Barnard, J. A. *Flame and combustion*, 3rd ed., Glasgow: Blackie Academic and Professional, 1995.

[17] Gupta, Ashwani K. and Lilley, David G. Combustion and environmental challenges for gas turbines in the 1990s. IN: *Journal of Propulsion and Power*, vol. 10, no. 2, 1994, 137-147.

[18] Hansen, A. and Kappis, W. Automised Calibration of Empirical Loss- and Deviation Models for Compressor Blade Rows. ASME, 2001-GT-0346. IN: *Proceedings of the 2001 International Gas Turbine and Aeroengine Congress & Exhibition*, 4-7 June 2001, New Orleans, USA.

[19] Harper, I. *RB211-524G/H Boeing 747 and 767 In Flight Relighting Summary.* Rolls-Royce, MDR69626, 1990.

[20] Holdeman, J. D., Mongia, H. C. and Mularz, E. J. *Assessment, development, and application of combustor aerothermal models.* NASA, NASA-TM-100290, 1988.

[21] Howell, A. R. *The present basis of axial flow compressor design.* RAE/E-3946. HMSO, 1942.

[22] Howell, A. R. *The windmilling of compressors in turbo-rocket engines.* NGTE/M-267. NGTE, 1956.

[23] Howell, A. R. and Carter, A. D. S. *Fluid flow through cascades of aerofoils.* NGTE/PJ/R-6. NGTE, 1946.

[24] Jones, J. C. *Combustion science – principles and practice.* Newtown, Australia: Millennium Books, 1993.

[25] Koblish, T. R., Patel, S. G. and Siddiqui, J. R. Fuel spray characteristics of gas turbine fuel injectors at cold start conditions. ASME, 95-GT-247. IN: *International gas turbine and aeroengine congress and exposition*, 5-8 June 1995, Houston, Texas, USA.

[26] Lee, F. P. and Koblish, T. Numerical correlation of gas turbine combustor ignition. ASME, 88-GT-242. IN: *Gas turbine and aeroengine congress and exposition*, 6-9 June 1988, Amsterdam, Netherlands.

[27] Lefebvre, A. H. *Gas turbine combustor design problems.* Washington D.C.: Hemisphere Publishing Corp., 1980.

[28] Lefebvre, A. H. Ignition theory and its application to the altitude relighting performance of gas turbine combustors. IN: *Combustion and heat transfer in gas turbine systems, Proceedings of an International Propulsion Symposium*, 15-17 April 1969, College of Aeronautics, Cranfield, UK, vol. 11, 105-119.

[29] MacCallum, N. R. L. Effect of bulk heat transfer in aircraft gas turbines on compressor surge margins. IN: I. Mech. E. *Heat and fluid flow in steam and gas turbine plant.* London, 1974, 94-100.

[30] MacCallum, N. R. L. and Grant, A. D. *The effect of boundary layer changes due to transient heat transfer on the performance of an axial-flow air compressor.* SAE 770284, 1977.

[31] Morita, M., Sasaki, M. and Torisaki, T. Restart characteristics of turbofan engines. IN: AIAA, *International Symposium on Air Breathing Engines, 9th*, vol. 2, 3-8 September 1989, Athens, Greece, 1200-1206.

[32] Neal, P. F. Mechanical and thermal effects on transient and steady state component performances of gas turbines. AGARD, CP-324, 1982.

[33] Oates, G. C. The aerothermodynamics of aircraft gas turbine engines. AFAPL-TR-78-52, 1978.

[34] Perez-Blanco, Horacio and Henricks, Todd B. Gas turbine dynamic model for simulation and control. ASME, 98-GT-78. IN: *Proceedings of the 1998 International Gas Turbine & Aeroengine Congress & Exhibition*, 2-6 June 1998, Stockholm, Sweden.

[35] Rodgers, C. Starting torque characteristics of small aircraft gas turbines and APU's, IN: ASME, *Journal of Engineering for Power*, Vol. 102, April 1980, 231-238.

[36] Rosfjord, Thomas J. and Cohen, Jeffrey M. Evaluation of the transient operation of advanced gas turbine combustors. IN: AIAA, *Journal of Propulsion and Power*, Vol. 11, No. 3, 1995, 497-504.

[37] Ryan, J. P. *Compressor performance representation using derived stage characteristics.* MSc thesis, Cranfield University, August 1999.

[38] Sanborn, J. W., Lenertz, J. E. and Johnson, J. D. Advanced turbofan engine combustion system design and test. IN: *AIAA, SAE, ASME and ASEE, Joint propulsion conference, 23rd*. 1987.

[39] Schultz, Donald F. and Mularz, Edward J. *Factors affecting altitude relight performance of a double-annular ram-induction combustor.* NASA, TM X-2630, 1972.

[40] Schultz, Donald F. *Variable combustor geometry for improving the altitude relight capability of a double annular combustor.* NASA, TM X-3163, 1975.

[41] Steele, L. L., Anex, R. P., Grant, J. R., Harrold, D. P., Erhart, J. J. and Rock, S. M. *Application of system identification techniques to combustor post-stall dynamics, final report.* Pratt and Whitney Aircraft, West Palm Beach, FL. Government Products Div., AD-A178082, 1987.

[42] Wells, R. G. *The transient performance simulation of an industrial gas turbine.* M.Sc. Thesis, Cranfield Institute of Technology, 1985.

[43] Xiong, T.-Y., Huang, Z.-X. and Wang, Y.-Z.. Some factors affecting altitude relighting performance of turbojet engine combustor. ASME, 81-GT-49, 1981.

[44] Zhao, Q. S. Calculation of windmilling characteristics of turbojet engines. ASME, 80-GT-50. IN: *Gas Turbine Conference and Products Show*, 10-13 March 1980, New Orleans, LA, USA. (Also: ASME, *Journal of Propulsion and Power*, Vol. 103, Jan. 1981, 1-12.)